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HANDBOOK OF ACOUSTIC NOISE CONTROL
Volume 1. Physical Acoustics

EDITORS

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NOTICE

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FOREWORD

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A B S T R A C T

This handbook, comprising two volumes, is intended to provide an overall view of the noise control problem. Classically, the word noise denotes a phenomenon related to hearing, and noise is commonly defined as any unwanted sound. In recent years, noise has come to mean the antithesis of desired signal in any stimulus or form of energy. Thermal motions produce noise in an electronic system and ground reflections may constitute noise in a radar device. Thus it becomes desirable to designate explicitly the subject matter of this handbook, acoustic noise.

There are several ways in which acoustic noise can be undesirable; it can produce pain and damage of personnel, it can interfere with speech communication, and it can cause annoyance and general degradation of environment for work and relaxation. These matters are the subject of Volume II, in which the several subjective responses are analyzed and correlated with properties of the physical stimuli. Volume I is concerned with the stimuli themselves, with their generation and with their control.

However, the problems encountered in the design of a noise control installation are rarely of a purely physical or biological nature, but include the varied factors of economics, operations and planning. Thus the acoustical engineer must be capable of effecting a compromise between these often contradictory considerations. The present volume, therefore, is a guide to assist in making a rational approach to the problems of noise control.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDING GENERAL:

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PART I. INTRODUCTION

CHAPTER 1

GENERAL ASPECTS OF NOISE CONTROL

1.1 Some Basic Concepts

Problems of noise control arise because acoustic noise interacts with man and even with machines. Psychologically speaking, noise is a stimulus, to which man reacts with a response. The nature of the response is dependent upon a wide variety of factors. The response may consist of a negligible reaction, as is often the case with a noise of low intensity or one to which the listener has become accustomed and adjusted. With increasing intensity of the noise stimulus, the response generally progresses through increasing degrees of annoyance, until conditions are reached in which pain or damage may also occur. In some instances man's response to noise is expressed in terms of the effect upon certain normal human activities. For example, the response to noise may be described in terms of interference with speech communication, or in terms of interference with sleep.

Because the human response to sound involves inherent variability, characteristic of all biological organisms, acoustics has sometimes been regarded as a field which lacks the precision that is characteristic of other branches of engineering and science. It is indeed true that general conclusions concerning man's response to sound do not usually apply accurately to a particular individual, and it is sometimes true that high precision is not attainable in the physical measurement of noise. However, the use of statistical concepts, in the numerical description of the important physical and psychological variables, has made these difficulties less serious and has helped to place the field of noise control on a more precise scientific basis.

The need for statistical description arises in connection with both the stimulus and the response. For example, the noise resulting from automobile traffic, or from overhead airplane traffic, is by no means uniform. The stimulus in these cases may be described in terms of the average number of occurrences per hour, and the average characteristics of the noise for each occurrence.

At the same time, there exist significant differences in the responses of people to a stimulus. Some residents will complain violently about a given noise, whereas others, at the same time, will be only mildly annoyed, or will not express annoyance. The human reaction, then, must be specified in such terms as the "likelihood of a given response", or the percentage of people who are annoyed. Even though these figures do not necessarily describe the response of a given individual, the results can describe with accuracy the collective or average response of a community.

When adequate engineering descriptions have been established for both the noise stimulus and man's likely response (to be expected to the stimulus), the acoustical engineer is in a position to prepare quantitative recommendations for noise-control measures which will modify the stimulus intensity and characteristics in such a way as to render any specified response unlikely. This is the essence of the noise-control problem. The important aspects of noise control can be described under three headings, as indicated below.

1. Specification of the noise source. This aspect of noise control has to do with purely physical measurements of the amount of sound power produced by the source and the manner in which the power is distributed.

2. Specification of allowable noise levels. This is largely a bio-acoustical aspect of the noise control problem. Allowable noise levels are engineering recommendations formulated from (a) the observed responses of man to the stimulus; (b) policy judgements on the acceptable degree of response consistent with economic and operational factors.

3. Design of noise-control components. The available engineering information on noise reduction produced by various structures is utilized, to select control measures which will insure that noise levels will not exceed the allowable values.

The present volume of this manual is concerned with the purely physical aspects of the noise-control problem, that is, with the properties of noise sources and with the properties of noise-control components. The problem of establishing the bio-acoustic basis for determining allowable noise levels is discussed in Volume II of this manual.

1.2 Specification of a Noise Source

In order to give a quantitative description of a noise source, it is necessary to have equipment capable of measuring the strength of an acoustical signal, suitable techniques for the use of this equipment, and appropriate terminology and conventions for describing the results. This is the subject matter of Chapter 2. Usually the acoustical measuring equipment includes a pressure-sensitive microphone, an amplifier, an adjustable electrical "filter" to restrict the range of frequencies to which the equipment is responsive, and a meter or other device which gives an indication of the strength of the acoustical signal within the selected range of frequency. When this equipment is operated properly at a specified location, the resulting data consist of (1) a tabulation or graph of the sound pressure distributed in specified frequency bands; (2) the "overall" sound pressure, which is measured when the equipment is made responsive to all audible frequencies simultaneously. Sometimes, in a preliminary survey, only the second quantity is measured. Actual data are reported in terms of <u>sound pressure level</u>, a quantity which is related to the logarithm of the sound pressure and which is described in units of <u>decibels</u>.

In a very few noise-control problems, an analysis of sound pressure at a single location may give sufficient design information. More frequently, however, complete specification of the noise source is required. A complete specification consists of stating (1) the acoustical power developed by the source, and (2) the directional distribution of the radiated power. Both pieces of information must be given with respect to (3) the frequency distribution.

The methods for obtaining a complete specification of a noise source are considered in Chapter 3. These methods require sound pressure measurements to be made at a number of locations about the source. In situations where the source operates intermittently, it may be necessary to specify the temporal pattern of operation in addition to the acoustical properties of the source.

A sound survey which will lead to complete specification of a noise source is costly and time-consuming. Moreover, it is often desirable to plan noise-control components before the noise source is installed and operated. Therefore, it is advantageous for the sound-control engineer to have available the results of past measurements on noise sources in various

categories. When the past measurements for a noise-producing device of one type are sufficiently extensive, that information may be used to construct design charts, from which the noise-source characteristics of any particular device of the same type may be predicted. Such design charts relate the noise-producing properties of the device to certain operating parameters. For example, noise produced by an aircraft propeller is related to driving horsepower per blade and to tip speed. Chapters 4-8 are devoted to a presentation of accumulated data for noise sources of several types. Where possible, the information is reduced to design charts.

1.3 Methods of Noise Control

When the noise source has been adequately specified, and when a decision has been reached as to the maximum allowable noise levels at the points of observation, the acoustical engineer then combines these pieces of information to formulate recommendations and design specifications for the most economical noise-control system. The general principles of this procedure are discussed in Chapter 9. The specific calculations which lead to acoustical specifications for the sound-control components are discussed in Chapter 10.

When the numerical performance specifications for the noise-control components (walls, mufflers, etc.) have been derived, detailed structural designs for the components are prepared. The noise-reducing components must meet the specifications, but must not exceed specifications to a wasteful degree. The detailed designs which will meet specifications properly can be prepared only on a basis of quantitative theoretical and experimental performance data. Chapters 11-13 contain performance data for a variety of noise-control components, and also information upon the reduction of sound intensity by distance and by directional radiation.

1.4 Evaluation

After noise-control components have been installed, it is advisable to make a quantitative evaluation of the effectiveness of the entire noise-control system. Evaluation measurements are desirable, (1) to ascertain whether the noise-reduction requirements have been met; (2) to determine whether individual sound-control components are functioning

properly; (3) to provide engineering data for future use. Evaluation procedures require a series of sound-pressure measurements after the installation has been put in operation. Evaluation methods are discussed in Chapter 14.

CHAPTER 2

TERMINOLOGY AND MEASUREMENTS

2.1 Definitions

The following terms are used frequently and so their definitions are included here. For a more inclusive list and for a more extended discussion, the reader is referred to the sources listed in Sec. 2.5, especially:

American Standards Association Bulletin on Acoustical Terminology

Beranek, L. L. - Acoustic Measurements, J. Wiley and Sons, 1949, Chapter 1.

Absorption coefficient for a surface is the ratio of the sound energy absorbed by a surface of a medium (or material) exposed to a sound field (or to sound radiation), to the sound energy incident on the surface. The stated values of this ratio are to hold for an infinite area of the surface. The conditions under which the measurements of the absorption coefficient are made are to be stated explicitly with all reported values.

Three types of absorption coefficients associated with three methods of measurement are:

Chamber absorption coefficient is obtained from measurements made in a reverberation chamber. The conditions under which the coefficient is measured including pertinent data on the chamber itself are to be stated explicitly.

Free-wave absorption coefficient is obtained from measurements made with a plane, progressive, sound wave incident on the surface of the medium. The angle of incidence is to be stated explicitly. When the angle of incidence is normal to the surface, the coefficient is called normal free-wave absorption coefficient.

Sabine absorption coefficient is obtained from the integration of free wave absorption coefficients over all the angles of incidence.

Articulation testing is a procedure by which a quantitative measure of the intelligibility of speech is obtained.

Articulation score is the percentage of speech items correctly recorded by a group of listeners who are listening to sentences, words, or syllables read by a talker. This term can be specialized depending on the type of test used into:

> Sound articulation score is the articulation score when the items are fundamental speech sounds when the sounds are combined into meaningless syllables.
>
> Syllable articulation score is the articulation score obtained when the speech items are syllables.
>
> Word articulation score is the articulation score obtained when the speech items are words.
>
> Sentence articulation score is the articulation score obtained when the speech items are sentences.

Attenuation is the reduction expressed in decibels, of the sound intensity at a designated first location as compared to sound intensity at a second location which is acoustically farther from the source. For an acoustic signal which travels in open air, the observation points must lie on the same radial line from the source. For a signal confined to a channel or duct, the intensity values are averages over the cross-section. Reflected signals are disregarded in computing the attenuation.

Attenuation constant is the real part of the propagation constant. The unit is the neper per unit distance.

Beats are the periodic amplitude variations resulting from the addition of two or more waves of different frequencies.

Decibel (db). The decibel is a dimensionless unit for expressing the ratio of two powers. The number of decibels is 10 times the logarithm to the base 10 of the power ratio. With W_1 and W_2 designating two powers, or, in acoustics, two intensities, and n the number of decibels corresponding to their ratio,

$$n \text{ (in db)} = 10 \log_{10} \frac{W_1}{W_2} .$$

When the impedances are such that ratios of currents, voltages, pressures, or particle velocities are the square roots of the corresponding power ratios or intensity ratios, the number of decibels by which the corresponding powers or intensities differ is expressed by:

$$n = 20 \log_{10} \frac{i_1}{i_2}$$

$$n = 20 \log_{10} \frac{V_1}{V_2}$$

$$n = 20 \log_{10} \frac{v_1}{v_2}$$

$$n = 20 \log_{10} \frac{p_1}{p_2}$$

where i_1/i_2, V_1/V_2, v_1/v_2, and p_1/p_2 are the given current, voltage, velocity and pressure ratios, respectively.

Diffraction. See the definition of scattering.

Diffuse sound exists when the energy density is uniform in the region considered and when all directions of energy flux at all parts of the region are equally probable.

Directivity factor is the ratio of the mean-square pressure (or intensity) on a designated axis of a transducer at a stated distance to the mean-square pressure (or intensity) that would be produced at the same position by a spherical source if it were radiating the same total acoustic power. A free field is assumed as the environment. The point of observation must be sufficiently remote from the transducer for spherical waves to exist.

Directivity index is ten times the logarithm to the base 10 of the directivity factor. The unit is the decibel.

Energy density at a point is the energy per unit volume in a sound wave. The unit is the erg per cubic centimeter.

Energy flux of a sound field is the average, over one period or a time long compared to a period, of the rate of flow of sound energy through any specified area in a direction perpendicular to that area. The unit is the erg per second. Expressed mathematically, the sound energy flux J is

$$J = \frac{1}{T}\int_0^T pSv_a dt$$

where T is the period or time long compared to a period,

p is the instantaneous sound pressure over the area S,

S is the area,

v_a is the component of the instantaneous particle velocity in the direction normal to the area S, i.e.,

$$v_a = v \cos \theta$$

where v is the instantaneous particle velocity,

θ is the angle between the direction of propagation of the sound and the normal to the area S.

Forced vibration is a vibration directly maintained in a system by a periodic force and having the frequency (or frequencies) of the force.

Free field is a sound field in an isotropic, homogeneous region free from bounding surfaces.

Free wave (free progressive wave) is a sound wave propagated in a medium free from boundary effects. A free wave in a steady state can only be approximated in practice.

Frequency (in cycles per second, or cps) is the rate of repetition of a periodic phenomenon. The frequency is the reciprocal of the period, or the time necessary for the phenomenon to repeat. The frequency (f) of a sound wave is equal to the ratio of the speed of sound (c) to the wave-length of sound(λ), $c = f\lambda$.

Hearing loss (deafness) of an ear at a specified frequency is the difference in decibels between the threshold of audibility for that ear and the normal threshold of audibility at the same frequency.

Impedance

Specific Acoustic Impedance at a point is the complex ratio of the sound pressure to the particle velocity.

Intensity (sound energy flux per unit area) at a point in a specified direction is the sound energy transmitted per unit of time in the specified direction through a unit area normal to this direction. The unit is the erg per second per square centimeter, or watts per square centimeter. Expressed mathematically, the sound intensity I is:

$$I = \frac{1}{T}\int_0^T pv_a \, dt$$

where T is the period or a time long compared to a period,

p is the instantaneous sound pressure at the point,

v_a is the component of the instantaneous particle velocity in the specified direction, i.e., $v_a = v \cos \theta$

where

v is the instantaneous particle velocity,

θ is the angle between the direction of propagation of the sound and the specified direction.

In a gas of density ρ_0, for a plane or spherical free wave having a velocity of propagation c, the sound intensity corresponding to a root-mean-square sound pressure p_e is

$$I = \frac{p_e^2}{\rho_0 c} \cos \theta \quad \text{ergs/sec/cm}^2 .$$

Level

Intensity level, in decibels, of a sound is 10 times the logarithm to the base 10 of the ratio of the intensity I of this sound to the reference intensity I_0. The reference intensity I_0 must be stated and is usually 10^{-16} watts/cm^2. Conventional sound pressure meters and sound level meters do not measure intensity. Hence, the words "intensity level" should not be applied to data taken with them. Instead, the two expressions "sound level" and "sound pressure level" are used in this manual.

Loudness level of a sound is numerically equal to the sound pressure level in decibels of the 1000-cycle pure tone which is judged to be of equal loudness. The unit is the phon.

Noise level (in decibels) is the term which for convenience is applied to a noise to specify its "sound level" as defined above.

Power level is the ratio, expressed in decibels, of the total acoustic power W radiated by a sound source to a reference power P_{ref}. Thus:

$$PWL = 10 \log_{10} (W/P_{ref}).$$

Three values of P_{ref} have been used. These are 0.90×10^{-13} watt, 0.93×10^{-13} watt, and 1.00×10^{-13} watt. For convenience, the reference of 1.00×10^{-13} watt is used in this manual. The maximum spread in the calculated power level caused by using one or another of the three values is less than 0.5 db, which is less than the uncertainty of most noise measurements. The power level charts given in this manual for various noise sources, therefore, can be used without correction with any of the three reference levels. When any of the above reference levels is used, the power level is approximately equal to the sound pressure level in air at a distance of 0.282 ft from the effective center of a non-directional source. In this case the energy output of the source is uniformly distributed over an area of one sq ft. See Chapter 3 for a more detailed discussion.

Sound level at a point is the weighted sound pressure level determined by instruments which meet the specifications drawn up by the American Standards Association for sound level meters.

Sound Pressure level in decibels of a sound is 20 times the logarithm to the base 10 of the ratio of the rms sound pressure at that point to a reference pressure. The reference pressure is usually 0.0002 dyne/cm^2 and must be stated in all cases.

Spectrum level at a specified frequency is the sound pressure within a band one cps wide centered at the frequency. The unit is the decibel.

Loudness is that aspect of auditory sensation in terms of which sounds may be ordered on a scale running from "soft" to "loud". Loudness is chiefly a function of the pressure amplitude of a sound, but it is also dependent on the frequency composition. The unit is the sone. The quantitative measurements of this subjective sensation depend on a statistical analysis of tests made on a great many people. (See Volume II for discussion)

Masking is the number of decibels by which a listener's threshold of audibility for a given tone is raised by the presence of another sound. This masking depends on both the pressure amplitude and the frequency composition of the masking sound, and also on the pressure amplitude and the frequency of the tone which is masked.

Microbar is a unit of pressure commonly used in acoustics. One microbar is equal to one dyne per square centimeter. The use of the word bar instead of microbar to designate this quantity is disapproved.

Neper is a dimensionless unit for expressing the ratio of two powers. The number of nepers is 1/2 the natural logarithm of the power ratio. One neper is equal to 8.686 db.

Noise is any undesired sound. As used broadly in acoustics, this may include not only industrial sounds such as traffic and machinery, but also speech and musical sounds if they are undesired at particular locations.

> Random noise is a sound or electrical signal whose amplitude has instantaneous values which are distributed according to a normal (Gaussian) curve. Random noise can have any frequency spectrum.

Noise reduction is the decrease of the sound pressure level at a specified observation point which is attributable to a designated structure. Noise reduction is also used to designate the difference in the sound pressure levels existing at two different locations at a single time, when designated structures are in position. The term Noise Reduction is meaningful only when the noise-control components and the points of observation are fully specified.

Normal threshold of audibility is for any frequency that arbitrary value of sound pressure level which is produced by an approved audiometer at the "zero db" setting. This value represents approximately the threshold of hearing for young persons having no hearing damage.

Octave is the interval between any two frequencies whose frequency ratio is 2:1.

Particle velocity. In a sound wave the particle velocity is the instantaneous velocity of a given infinitesimal part of the medium, with reference to the medium as a whole, due to the passage of the sound wave. The unit is centimeters per second.

Pitch is that aspect of auditory sensation in terms of which sounds may be ordered in a scale running from "low" to "high". Pitch is chiefly a function of the frequency composition of a sound, but it is also dependent on the pressure amplitude. The unit is the mel. For a graphical relationship between pitch in mels and frequency in cps, which has been established for pure tones, see Fig. 5.17 of Acoustic Measurements by Beranek.

Pressure

Average sound pressure is the value of the time integral over a period or a time long compared to a period of a rectified sound wave. The unit is the microbar.

Barometric pressure (static or ambient pressure) is the pressure that would exist in the medium if no sound wave were present. The unit is the microbar.

Effective sound pressure at a point is the root-mean-square value of the instantaneous sound pressure, taken over a complete cycle or a period long compared to a cycle, at that point. The unit is the microbar.

Instantaneous sound pressure at a point in the total instantaneous pressure at that point minus the static pressure. This quantity is often called excess pressure. The unit is the microbar.

Maximum sound pressure for any given cycle is the maximum absolute value of the instantaneous sound pressure during that cycle. The unit is the microbar. In a sinusoidal sound wave this maximum sound pressure is also called the pressure amplitude.

Propagation constant of a plane sound wave in a homogeneous infinite medium is the natural logarithm of the ratio of the steady-state velocities or pressures at two points separated by unit distance in the medium. The ratio is determined by dividing the value of the velocity (or pressure) at the point nearer the transmitting end by the value of the velocity (or pressure) at the point more remote. Single frequency pressures and velocities are here supposed to be represented by complex numbers. Their ratio is therefore a complex number. The real and imaginary parts of the propagation constant are the attenuation constant and phase constant respectively. The units of these two quantities are nepers and radians, respectively, per unit distance.

Rayl is a unit of specific acoustical impedance. A specific acoustic resistance, reactance, or impedance is said to have a magnitude of one rayl when a sound pressure of one dyne/cm^2 produces a linear velocity of one cm/sec.

Response is the ratio of some measure of the output of a device to some measure of the input, under conditions which must be stated explicitly. The response characteristic, often presented graphically, gives the response as a function of some independent variable such as frequency.

Reverberation is the persistence of sound in an enclosure after the radiation from the original source has ceased. As distinguished from discrete echoes, reverberant sound contains a large number of reflected components which blend together.

Reverberation chamber. An enclosure whose boundaries are highly reflecting acoustically.

Reverberation time for a given frequency is the time required for the average sound energy density, originally in a steady state, to decrease after the source is stopped, to one-millionth of its initial value (60 db).

Room factor is the name given the expression S ln $(1/1-\bar{\alpha})$ where S is the area of the room in sq ft and $\bar{\alpha}$ is the weighted mean of the absorption present in the room, i.e.

$$\bar{\alpha} = \frac{1}{S}\left(S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + \ldots\ldots + S_n\alpha_n\right).$$

Root-mean-square (rms) value of a varying quantity is the square root of the mean value of the squares of the instantaneous values of the quantity. In periodic variation the mean is taken over one period.

Sabin is a unit of absorption. It is the equivalent of one sq ft of a perfectly absorptive surface.

Scattering of sound refers to the changes in a radiation field caused by the introduction of a material object or inhomogeneities in the propagation media. Thus if in the absence of the above objects or inhomogeneities, the radiation field is described by a wave function ψ_I and with the objects present by a new wave function ψ, then

$$\psi = \psi_I + \psi_{II}$$

where ψ_{II} is called the "scattered" wave. In some cases, the wave function ψ_{II} is called the "diffracted" wave although there is no physical distinction to be drawn between the two words.

Sensitivity

Free-field sensitivity of a microphone is the ratio of the electrical output, measured in a specified manner, to the free-field sound pressure existing at the microphone location prior to the introduction of the microphone in the sound field. The free-field sensitivity is defined for a

plane progressive sound wave whose direction of propagation has a specified orientation with respect to the principal axis of the microphone.

Axial sensitivity of a microphone about an axis perpendicular to the diaphragm is the free-field sensitivity when the sound is incident normally on the diaphragm.

Simple source of sound is a source which radiates sound uniformly in all directions under free-field conditions.

Sound, objective or physical is an alteration of pressure, stress, particle displacement, or particle velocity which is propagated in an elastic material or the superposition of such propagated alterations.

Sound, subjective is that auditory sensation which is evoked by objective or physical sound.

Spectrum is a distribution as a function of frequency of the components of a wave.

Continuous spectrum is a spectrum the components of which are continuously distributed over a frequency range.

Line spectrum is a spectrum the components of which are confined to a number of discrete frequencies.

Power spectrum is the relative distribution in frequency at a point in space of the acoustical energy radiated by a sound wave.

Power spectrum level at a specified frequency is the power level of a band one cps wide centered at the frequency. The unit is the decibel.

Standing waves result from the interference of progressive waves of the same frequency.

Steady state. A system is said to have reached a steady state condition when the relevant variables of the system no longer change as a function of time. See definition of transient.

Transducer. A transducer is a device which receives energy from one or more systems and delivers the energy in different form, to one or more other systems.

Electroacoustic transducer is a transducer which is actuated by power from an electrical system and supplies power to an acoustical system or vice-versa.

Transient. A system is said to be in a transient state if the relevant variables of the system are varying with time. When the transient state no longer exists, it is said that steady-state conditions have been reached.

Transmission coefficient is the ratio of the energy transmitted through an interface or septum between two media to the energy incident on the interface or septum.

Transmission loss is the ratio, expressed in decibels, of the sound energy incident on a structure to the sound energy which is transmitted. The term is applied both to building structures (walls, floors, etc.) and to air passages (mufflers, ducts, etc.).

Vibration is an oscillating displacement of a portion of a physical medium from a mean position. Objective sound, as defined above, is a special case of vibration. In architectural acoustics, the term vibration is usually applied to disturbances in solid structures (walls, floors, machines, etc.).

Ultrasonics is the general subject of sound whose frequency is greater than the highest frequency to which the ear can respond. The ultrasonic region can be considered to begin between 15,000 and 20,000 cps. The term supersonic is now disapproved because of its use in aerodynamics to mean a velocity greater than the velocity of sound.

Abbreviations

GLR - Graphic Level Recorder

SLM - Sound Level Meter

CRO - Cathode Ray Oscilloscope

OBA - Octave Band Analyzer

SIL - Speech Interference Level
(do not confuse with Sound Intensity Level)

SPL - Sound Pressure

PWL - Power Level

TL - Transmission Loss

NR - Noise Reduction

db - decibel

IL - Intensity Level

2.2 Statistics: An Introduction

Statistical methods are useful whenever there is available a large body of data which have been obtained by a repetitive procedure under essentially the same conditions. For example, if one wishes to know the diameter of a certain shaft with some accuracy, one can measure that diameter with a micrometer one hundred times. The repetition here is obvious, and one would resort to a statistical analysis of the data such as will be considered below to obtain one "best" value for the diameter. On the other hand, there seems to be no iterative procedure involved in compiling a table of the number of people who marry as a function of their age at marriage. While before the "experiment" consisted of measuring a given object, now each person represents one "experiment". Each "experiment" in the first case yielded a number -- a certain dimension. Likewise each of the second type of "experiment" yields a number -- the age at which that person marries. The iteration comes in canvassing a great many people to gather the data just as in the first case many measurements of the same type were made.

It is apparent to anyone who has ever made a measurement that repeating the measurement with the same instrument under what appears to be the same conditions does not, in general, yield the same result. If the results of a series of measurements are the same, then it means that the measuring instruement is too crude and one is not taking advantage of the full accuracy which the experiment is capable of yielding. But if the available accuracy is being pushed to the utmost, the multitude of small perturbations to which a measuring system is susceptible will conspire to produce slightly different results at each trial. Thus with a micrometer, the pressure of the barrel against the piece, the angle at which the scale is viewed, ambient temperature variation, etc., all produce a variation in the required result.

Imagine the following hypothetical situation. A diameter is measured with an accurate micrometer and this measurement is repeated an <u>infinite</u> number of times. Let each measurement be recorded as it is obtained. Then this collection contains within it every possible result which can ever be obtained by performing that experiment, simply because in repeating the experiment an infinite number of times every such result <u>was</u> obtained. Such a hypothetical collection of values is spoken of as the "infinite population" or the "parent population". If instead of performing such a herculean task, we measure something

only ten times, we say that we have "extracted a sample from the infinite population". Every one of the ten measurements is one of those within the far more inclusive infinite population. Because of our finite lifetime we could never come near to arriving at all the result contained within the parent population. This then immediately defines the goal of satistical analysis. Given a finite number of measurements which are a sample of an infinite population, to extract a maximum of information about that infinite population from which they were drawn.

The infinite number of measurements which are the parent population can be recorded in two ways: they can be left as entries in a notebook just as they were gathered in the hypothetical experiment which was proposed, or they can be presented in the form of a graph. For example, if one measured something of about one-half inch diameter with a micrometer which reads to three decimal places, the parent population is an infinite series of three digit numbers. Intuitively we know that these numbers are going to cluster around .500 in. and that certainly any one specific result, such as .497 in. will be obtained many times; and we have the feeling that a result like .365 in. will not come up very often. Therefore, one can present the data in the form of a histogram or bar graph. At each possible value a vertical column is drawn whose height is proportional to the frequency with which that value is obtained. Such a bar graph is called a frequency distribution. In the case of the diameter measurements, intuition and experiment lead to such a distribution as given in Fig. 2.1 (the actual choice of numbers will be explained later).

The distribution shown here is one of many possible frequency distributions and is known as a Normal or Gaussian distribution. There are two numbers necessary to specify such a distribution; the mean and the standard deviation. If these two numbers are known, then we know all there is to know about the distribution. In practice a sample is extracted from this population. This sample itself can be represented by a frequency distribution and this distribution should, in a general way., resemble that of the infinite population. The mean and standard deviation of the sample is calculated and it is hoped that the mean and standard deviation of the parent distribution, which is of course the only one in which we are really interested, will be about the same. An important result of statistical theory says that as the size of the sample increases, the mean of the sample approaches the mean of the parent population. Thus by

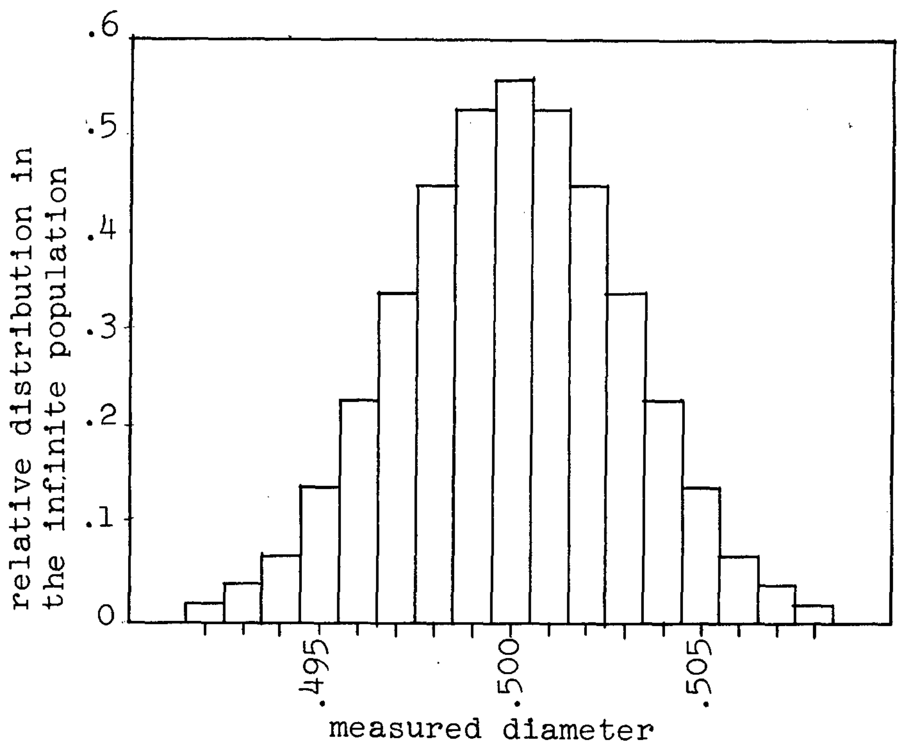


Figure 2.1

Distribution of measurements of the diameter of a circular shaft. This illustrates the shape of a Normal distribution.

taking enough measurements we can expect to get a reasonable picture of the parent population without actually going to the trouble of performing an infinite number of experiments. However, in practice one does not usually know ahead of time the form of the parent population.

Measures of Central Tendency. The quantities "mean" and "standard deviation" were mentioned above though these quantities were not defined. In order to summarize a series of results of the same experiment, one wishes to pick a representative value, i.e., some number about which all of the other numbers cluster. The most common measures of central tendency are the arithmetic mean, geometric mean, median, and mode. The arithmetic mean, or more simply, the mean is the common average. It is denoted by $\bar{x}$, and if the individual values are x_1, x_2, x_3 ......x_i.....x_n then

$$\bar{x} = \frac{1}{n} \sum_{i=1}^{n} x_i \quad .$$

If for some reason certain measurements are more reliable than others or are more important, weights w_1, $w_2 \ldots\ldots w_i \ldots\ldots w_n$ can be assigned each value and a <u>weighted mean</u> can be defined

$$\bar{x} = \frac{\sum_{i=1}^{n} w_i x_i}{\sum_{i=1}^{n} w_i} \quad .$$

This is the same as the mean of a distribution consisting of x_1 repeated w_1 times, x_2 repeated w_2 times, etc.

The <u>geometric mean</u> of n values of x is defined by

$$\bar{x} = \sqrt[n]{x_1 \; x_2 \; x_3 \ldots\ldots x_n}$$

It will be noted that this corresponds to averaging the logarithms of x_i. If the values x_i are listed in order of their magnitudes, then the middle value of such a listing, or the mean of the two middle values in case there are an even number of values, is defined to be the median. Finally, the value occurring most often is the <u>mode</u>.

To illustrate these definitions, consider the following distribution: 1, 2, 2, 3, 3, 3, 3, 3, 4, 5, 5, 6, 8. The arithmetic mean is 3.69, both mode and median are 3, and the geometric mean is 3.26.

There are two criteria for choosing a measure of central tendency:

1. which is more meaningful

2. which is more stable

To illustrate the first criterion look at the distribution of values as given in Fig. 2.2. The distribution is skewed toward the small values and in this case the mode is a better indication of central tendency than the mean, which is influenced by the tail of large values.

By stability is meant the measure of central tendency which would be most nearly repeated if the whole series of measurements were repeated. For example, suppose the average height of all

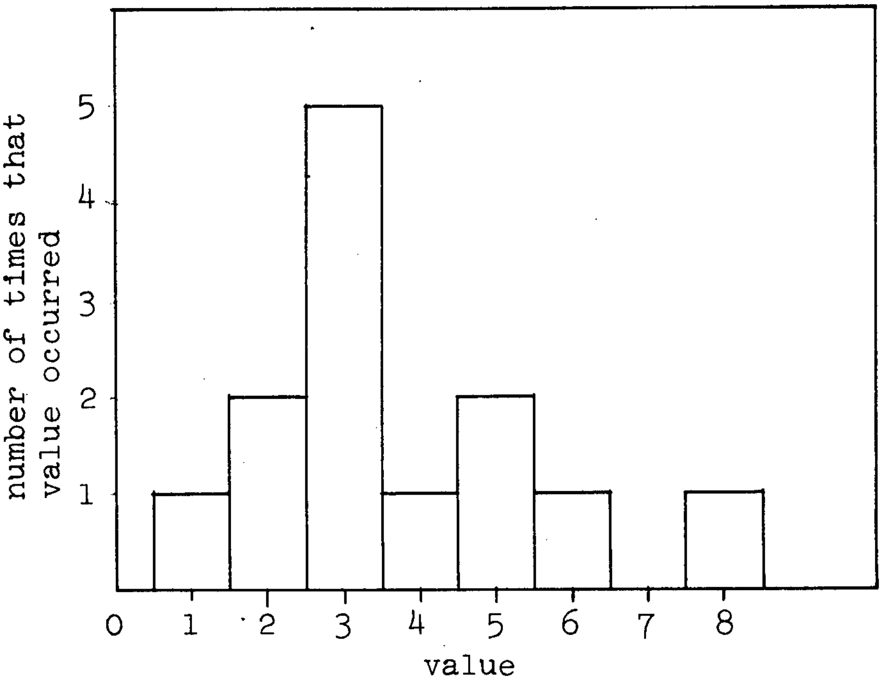


Figure 2.2

Arbitrary distribution of numbers for the purpose of illustrating the various measures of central tendency.

American males was desired. If one measured a group of ten men and found, for example, the mean and the median, and then continued this process with more groups of ten, one would find that the means of these small samples would be more tightly clustered than would the median. In such a case the mean is the better representative value because it is more stable.

The second physically important property of a distribution is the spread of variation which it possesses. Suppose two people each make a series of measurements in order to determine the weight of an object. If man A obtains measurements ranging from 4.8 to 5.3 lb and with a mean value of 5.1 and man B by more careful work finds the mean to be 5.0 from a series of values ranging from 4.9 to 5.1 lb, we are more apt to believe man B. He appears to be more reliable, more painstaking, and to allow fewer disturbing factors to enter into his work. Thus we see that the <u>range</u>, or difference between the largest and smallest values is significant in establishing the variation contained in a distribution.

There are three quantities used to indicate the same thing -- the variance, also called the second moment about the mean, the standard deviation, and probable error. Variance is defined by

$$s = \frac{\sum_{i=1}^{n} (\bar{x} - x_i)^2 w_i}{\sum_{i=1}^{n} w_i}$$

or

$$\frac{1}{n} \sum_{i=1}^{n} (\bar{x} - x_i)^2 .$$

It is the mean of the squared deviations from the mean. The standard deviation is the square root of the variance.

$$\sigma = \sqrt{s} .$$

The probable error is a poor measure of variation though it is used more than any other similar measure. This is possibly because it is the smallest of them all and therefore makes the mean look more reliable than would other measures. The probable error is defined as that deviation from the mean which will be exceeded 50% of the time. That is, if a result is reported 493 ± 5, then if the experiment were repeated again, there is a 50-50 chance that the measurement would differ from 493 by more than 5. Probable error can be found from the standard deviation, but only when the form of the frequency distribution is known. Thus if the distribution of errors is known to be Normal, then

$$\text{p.e.} = .674\ \sigma .$$

The relationship between probable error and standard deviation for a Poisson distribution approaches asymptotically the relation given for the Normal distribution as the mean of the Poisson distribution becomes large. Thus for a Poisson distribution for which $m = 1000$, p.e. $= .660\sigma$ while for $m = 20$, p.e. $= .575\sigma$. It is seen that using the relation p.e. $= .674\sigma$ will give a conservative estimate of the probable error. For small values of the mean, say less than ten, the probable error is a poor measure of the spread of a Poisson distribution; instead

the standard deviation should be used in such a case. While the standard deviation can be found from any given set of measurements, the use of probable error implies a knowledge of the error distribution. However, the first relation containing the factor .674 is of reasonably general applicability for the following reasons:

1. Normal error distributions are quite common.

2. A second common distribution, the Poisson, approaches the Normal distribution as the number of measurements is made large.

3. Even for a small number of measurements from a Poisson distribution, the factor .674 causes the probable error to be too large and hence is a more conservative estimate of error. This is to be desired.

It was mentioned above that two numbers, the mean and the standard deviation, are needed to characterize a Normal distribution. The Poisson distribution possesses the property that the standard deviation is related to the mean and so only one number is necessary to describe the distribution. The relation between the two is

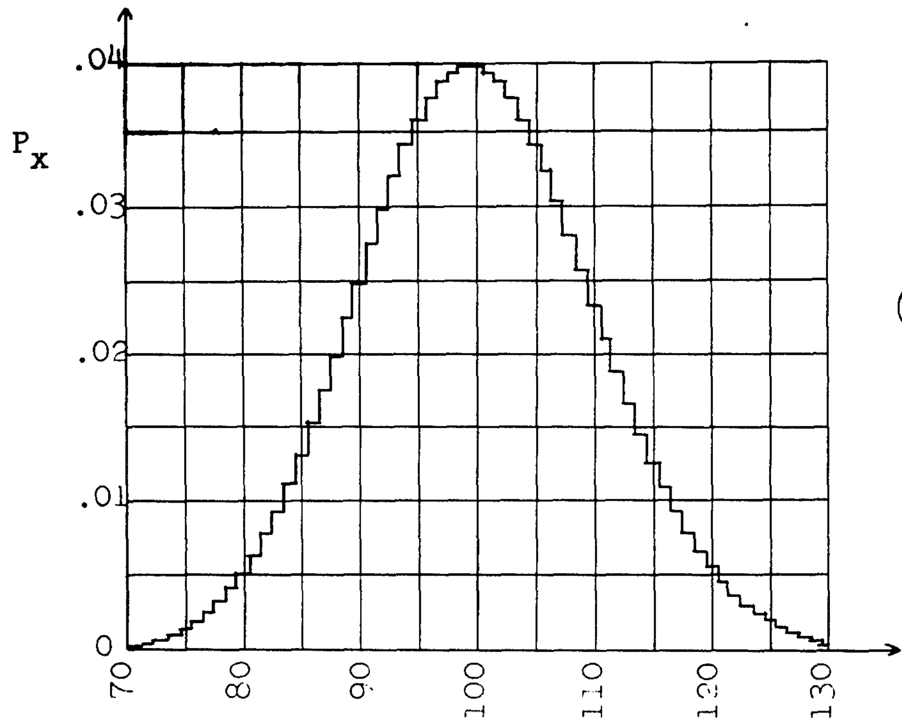
$$\sigma = \sqrt{\text{mean}} \qquad \text{Poisson}$$

The Poisson distribution is given by the mathematical expression

$$P_x = \frac{m^x e^{-m}}{x!}$$

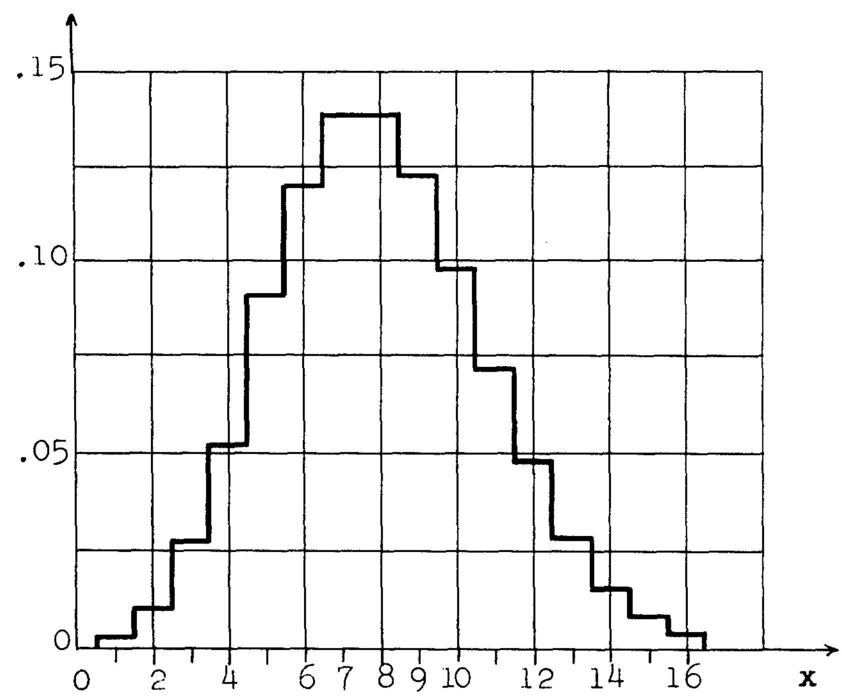
where m is the mean and P_x is the probability of finding a deviation of x from the mean. Figure 2.3 shows two Poisson distributions, one whose mean is 8, the second whose mean is 100. It will be noted that for m = 8 the distribution is decidedly skewed, but for m = 100 the skew is not noticeable. In fact it can be shown rigorously that as $m \to \infty$, the Poisson distribution approaches in shape the Normal distribution.

There is a distinction to be drawn between the terms "standard deviation of a sample" and "standard deviation of the mean of a sample". The first term relates to the deviation from the mean which might be expected if one more measurement were taken. The second indicates by how much a second mean would differ from the first mean, if the whole series of measurements were repeated.



P_x
.04
.03
.02
.01
0
70
80
90
100
110
120
130
x

(A)



.15
.10
.05
0
0
2
4
6 7 8 9 10
12
14
16
x

(B)

If σ is the standard deviation of a sample consisting of n measurements whose mean is x, then the standard deviation of the mean $\sigma_{\bar{x}}$ is

$$\sigma_{\bar{x}} = \frac{\sigma}{\sqrt{n - 1}}.$$

Such a relationship is intuitively acceptable. If fifty measurements are taken, one expects a certain amount of variation to be present, but averaging those fifty measurements gives one the feeling that the value is reasonably well established. If now someone else started over from the beginning and took fifty measurements, it would not be surprising to find his mean quite near the first mean. The above relationship states that a whole series of such means of fifty measurements should display only 1/7 the variation shown by the individual measurements which are combined to form any one mean.

Thus a mean value is to be reported as $\bar{x} \pm \sigma_{\bar{x}}$.

Whenever a measurement is made, it is necessary to make an estimate of its worth. This is to guide others who will use the measurement in judging its reliability. Thus there is a two-fold uncertainty in reporting any measurement: because a finite number of measurements must necessarily have been made, the mean of the measurements will differ by an unknown amount from the mean of the infinite population from which the measurements were drawn, and secondly the amount by which successive measurements can be expected to depart from the reported value can only be given as a but roughly-known probability. To say "It is probable that the value is-------" is meaningless unless the probability is given at least roughly in numbers. It may mean 90% or 99.99% probable. Yet from the very definition of probability this does not preclude the possibility of a very large deviation. One is only saying that such large deviations will not occur very often.

Propagation of Precision Indices. Having measured some physical quantities and attached to them some number indicating

---

Figure 2.3

(A) The normalized Poisson distribution having a mean of 8

(B) The normalized Poisson distribution having a mean of 100.

their reliability, it is usually necessary to use these numbers in computations. The question arises of what reliability is the computed quantity. For example, one may compute the acceleration of gravity g by measuring the length L and period T of a simple pendulum by knowing that

$$g = 4\,\pi^2\,\frac{L}{T^2}\;.$$

If X and Y are two measured quantities having standard deviations or probable errors (the same rules apply to both) of Δx and Δy respectively, then rules for the propagation of errors are:

For $z = x + y$, $z = x - y$ $\qquad \Delta z = \sqrt{(\Delta x)^2 + (\Delta y)^2}$

$z = x\,y$, $z = \frac{x}{y}$ $\qquad \frac{\Delta z}{z} = \sqrt{(\frac{\Delta x}{x})^2 + (\frac{\Delta y}{y})^2}$

$z = x^n$ (n integral or non-integral, positive or negative) $\qquad \frac{\Delta z}{z} = |n|\;\frac{\Delta x}{x}\;.$

It will be noted that $(\Delta x)/x$ is a relative error. An error of 1 ft in a measurement of 100 ft is 1% or a relative error of .01. Both the measurement and its precision index must of course be expressed in the same units.

Suppose, for example, that in the above case for the pendulum $L \pm \Delta L = 1.0003 \pm .0005$ m and $T \pm \Delta T = 2.005 \pm .003$ sec.

Then

$$g = 9.80 \text{ m/sec}^2$$

and

$$\frac{\Delta g}{g} = \sqrt{(\frac{\Delta L}{L})^2 + 2\,(\frac{\Delta T}{T})^2} = \sqrt{(\frac{5 \times 10^{-4}}{1})^2 + 2\,(\frac{3}{2 \times 10^3})^2}\;.$$

Hence $\Delta g = (22 \times 10^{-4})\,(9.8) = 216 \times 10^{-4} = .02 \text{ m/sec}^2$

and the result should be reported as

$$9.80 \pm .02 \text{ m/sec}^2\;.$$

There may arise more complex cases which these rules do not cover. Suppose $z = \sin x$ or in general $z = f(x)$. Then to a first approximation

$$\Delta z = \frac{\partial z}{\partial x} \Delta x \ .$$

For example, if $x = \frac{\pi}{4} (1 \pm .01)$ i.e., x is good to 1% and

$$z = \sin x \text{ then } \Delta z = \Delta x \cos s = .01 \quad (\tfrac{\pi}{4}) \frac{1}{\sqrt{2}} = .0071 \ (\tfrac{\pi}{4})$$

and z is known to .7%. In general, quantities raised to a power greater than 1 are known less accurately, relatively, than the original quantity while taking roots decreases the relative error.

Sampling. The previous discussions have all been concerned with large sampling. In order to get the best estimate of the mean of the parent distribution, many measurements have been taken in an effort to actually approach that infinite population. Great care must be taken to assure that such a sample is random. That is, there must be an equal probability of sampling any part of the parent distribution. If one wished to measure the distribution of frequencies in a certain noise sample, one would certainly not pass the noise through a high-pass filter first. With all the low frequencies missing or greatly reduced, the measurements would certainly indicate a preponderance of high frequencies. In fact, it is this inadvertent filter action of any instrument, electronic, optical, mechanical, etc., for which corrections must be applied.

While details will not be given, the existence of an extensive theory of small sampling should be mentioned. 1/ Large sampling attempts to discover the parameters of the parent population by a large number of measurements. As the number of measurements is made small, the values thus obtained become poorer and poorer estimates for what they are meant to be. What is needed then is a statistical theory which will take cognizance of the small size of the sample and will not even attempt to use it as being representative of the parent population. Thus the theory is inherently more advanced. One useful result of the theory of small sampling is the following: if from any parent population a series of small samples is extracted and their means are taken, these means will tend toward a normal distribution, regardless of the form of the parent population. If the mean of these means is taken and the series of measurements repeated to obtain more such grand means, these grand means will be even more normally distributed.

Inefficient Statistics. The statistical processes referred to previously have been termed "efficient" because they extract from a sample of an infinite population all possible information relating to that population. However, the application of such efficient statistics is apt to be time-consuming when there are many measurements to treat. Hence, in some cases it may be possible to extract from a given sample more information for a given amount of time or effort by the use of "inefficient" statistics. Application of such methods has the effect of ignoring some of the measurements for the sake of speed. To say that a method is 64% efficient means that the statistical quantities such as the mean, standard deviation, etc., are known about as accurately as they would be known if efficient methods were used on a sample from the same population which contained 0.64 as many measurements.

For example, suppose that a series of mean values are to be obtained, the experimental conditions being different for each mean value, and that the standard deviations for each mean value is to be smaller than a fixed value so that each point may be equally weighted. On would proceed to compute quickly after each measurement the standard deviation by inefficient methods, continuing this until enough measurements had been taken to meet the specifications. Or again, the mean of a set of measurements may be necessary to proceed with an experiment. In either of these cases speed rather than accuracy is desired.

Such techniques will be briefly stated.* References to the literature will be found at the end of the chapter. 2,3,4/ Rules and their efficiencies will be given for computing the mean and standard deviation of both large and small samples. In this case by large samples is meant greater than 100 values while small indicates 10 or less values. Patnaik 5/ has shown that for samples containing more than twelve measurements, the standard deviation may be estimated by averaging the results of treating the sample by breaking it down into groups of less than eight values.

* As given by R. D. Evans in a course at the Massachusetts Institute of Technology

1. Estimates of the mean of a large sample

    (a) The median is 64% efficient.

    (b) Averaging the values which are 29% of the way in from the upper and lower extremes of the data is 81% efficient.

    (c) The highest efficiency possible, 88%, can be obtained by averaging the values which are 20, 50, and 80% from the lowest value, i.e., the mid-value and the values 20% in from each end.

2. Estimate of the standard deviation of a large sample drawn from an approximately normal population

    (a) One-third of the difference between the values which are in 7% from each end is 65% efficient.

3. Estimates of the mean of a small sample $n < 10$

    (a) The median is 64% efficient.

    (b) For n = 3,5, the average of the largest and smallest values is more efficient, however. See Ref. (4).

    (c) For n = 7, 8, 9, 10 the average of the third term from each end is 84% efficient.

4. Estimate of the standard deviation of a small sample

    (a) The range divided by the square root of the number of values is 99% efficient for n = 3, falling to 85% for n = 10. See Ref. (5).

As an illustrative example in the use of inefficient statistics, consider the following situation. A person is measuring, under controlled laboratory conditions, the flow resistance * of a Fiberglas sample. The flow resistance is to be measured as a function of the density of the material, i.e., the same Fiberglas sample will be successively compressed and its flow resistance at various thicknesses will be determined. For greater accuracy, the flow resistance at any one thickness will be measured several times and the mean of the individual measurements taken. It is

* For a discussion of flow resistance measurements, see Acoustic Measurements by Beranek, p. 844.

desired that these means for each thickness all have the same weight. That is, the flow resistance at each thickness must be measured a sufficient number of times that the standard deviation of the measurements at that thickness is smaller than a certain fixed value.

Suppose that the measured values are as shown in the first column of Table 2.1. The question is, have enough measurements been made so that the standard deviation is, say 3%.

Tables 2.2 and 2.3 show the calculations necessary to compute the mean and standard deviation of the measured values, by both inefficient and efficient methods. Not only is the amount of work considerably lessened by the use of inefficient statistics, but the results are very close to the results obtained by the much longer efficient analysis.

TABLE 2.1

FLOW RESISTANCE OF A FIBERGLAS SAMPLE, 1 IN. THICK

Values, in the order of being measured	Values, listed according to their magnitude
34.6 rayls	31.4
32.5	32.0
31.4	32.5
33.5	33.0
32.0	33.2
33.7	33.5
34.7	33.7
33.2	34.6
33.0	34.7

TABLE 2.2

INEFFICIENT STATISTICS

$$\text{mean} = \frac{33.7 + 32.5}{2} = \frac{66.2}{2} = 33.1 \qquad \text{(See rule 3c)}$$

$$\text{S.D.} = \frac{\text{range}}{\sqrt{\text{No. meas.}}} = \frac{34.7 - 31.4}{\sqrt{9}} = \frac{3.3}{3} = 1.1 \qquad \text{(See 4a)}$$

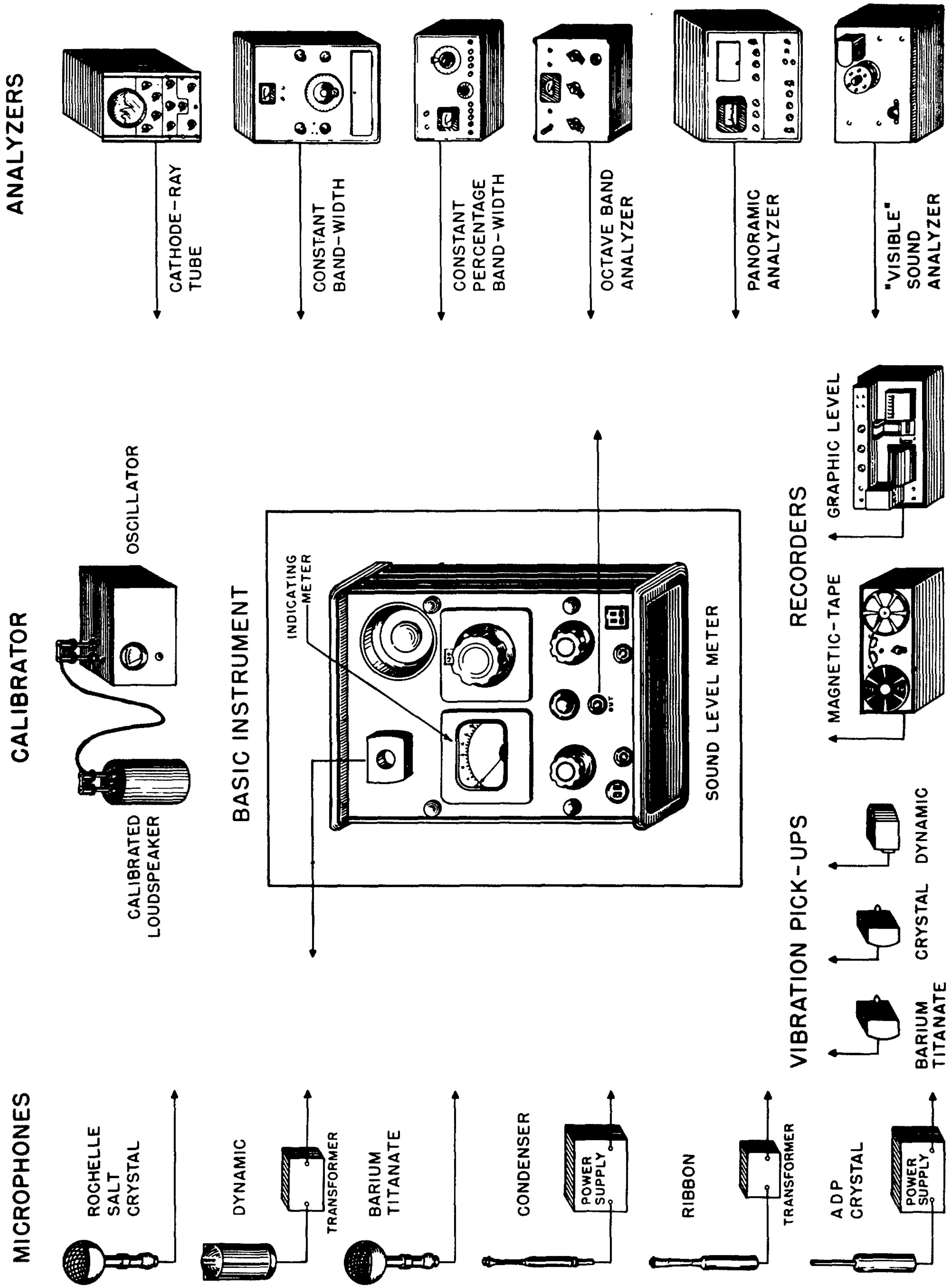
TABLE 2.3

EFFICIENT STATISTICS

measurement	abs. value of dev. from the mean	(deviation from mean)^2
34.6	1.4	1.96
32.5	.7	.49
31.4	1.8	3.25
33.5	.3	.09
32.0	1.2	1.44
33.7	.5	.25
34.7	1.5	2.25
33.2	0.0	0.00
33.0	.2	.04
9) 2728.6		9) 9.77
mean = 33.2		S.D. $= \sqrt{1.09} \approx 1.1$

The analysis shows that the standard deviation is 3.3%. In view of the inherent uncertainties in specifying statistical quantities (e.g., one can compute the standard deviation of the standard deviation) this point would be judged to have met the 3% criterion. Hence the experimenter would cease to make measurements at this thickness but would commence a series of measurements at some different thickness.

A note should be added at this point concerning the procedure to be used in recording experimental observations. It might be suggested that a convenient scheme when using inefficient statistics would be to note the values, leaving sufficient room between them to enable the writer to place the values in order of magnitude as they are obtained. This would be very poor practice and should never be done. A record of the values, in the order they were obtained, often gives valuable information on the behavior of the measuring system or the nature of the sampling process. For example, suppose that the flow resistance values were actually obtained in the order given in the second column of Table 2.1. One would certainly suspect that the high degree of ordering was not the result of randomness. A rising


MICROPHONES
ROCHELLE SALT CRYSTAL
DYNAMIC
TRANSFORMER
BARIUM TITANATE
CONDENSER
POWER SUPPLY
RIBBON
TRANSFORMER
ADP CRYSTAL
POWER SUPPLY
CALIBRATOR
CALIBRATED LOUDSPEAKER
OSCILLATOR
BASIC INSTRUMENT
INDICATING METER
SOUND LEVEL METER
VIBRATION PICK-UPS
BARIUM TITANATE
CRYSTAL
DYNAMIC
RECORDERS
MAGNETIC-TAPE
GRAPHIC LEVEL
ANALYZERS
CATHODE-RAY TUBE
CONSTANT BAND-WIDTH
CONSTANT PERCENTAGE BAND-WIDTH
OCTAVE BAND ANALYZER
PANORAMIC ANALYZER
"VISIBLE" SOUND ANALYZER

pressure in the air lines would be more likely and would indicate that none of the measurements are valid. Or in some experiments the process of measuring so disturbs the system that is being measured that it changes over to some other condition. The process of temperature measurements may actually change the temperature and the result which is obtained may be the equilibrium result after the system was disturbed. This is not the information which was desired.

2.3 Instruments

The apparatus most commonly used to measure the sound pressure levels and frequency spectrum of a sound field is shown schematically in Fig. 2.4. It consists of a microphone connected to a sound level meter, followed by some sort of electrical filter and either a recorder or a second meter. The SLM indicates the overall SPL. By means of an electrical filter or analyzer certain frequency bands can be selected and the portion of the total SPL contributed noise components by those bands can be measured individually by a second meter, which is usually built directly into the analyzer. Each of these components will be considered in greater detail below.

Microphones

In order to convert sound to electrical signals which are more readily measured, some sort of electro-mechanical transducer is necessary. There are two main types of microphones in use at audio frequencies (20 to 20,000 cps):

(a) Pressure microphones - responsive to instantaneous sound pressure.

(b) Velocity microphones - responsive to the fluctuating component of the instantaneous particle velocity.

Various types of microphones will be described below, chiefly from the point of view of their operating characteristics. For more complete, manufacturers' data on individual microphones or any of the standard texts 6,7/ in the field should be consulted.

Figure 2.4

The basic sound-measuring instrument, the sound level meter, and the various accessories commonly used in acoustic measurements (Photograph courtesy General Radio Company, Cambridge, Mass.).

(a) Pressure Microphones

Rochelle Salt - these microphones cover a frequency range of 20-8000 cps + 4 db, and have a dynamic range of from 20 to 160 db. They are sensitive to temperature changes because of the variation of the Rochelle salt dielectric constant with temperature. When the input impedance of the associated amplifier is high this is not serious except when considerable length of cable is used between the microphone and amplifier or sound level meter. In this case a temperature correction must be applied. The normal working temperature range is 10-100°F; the maximum safe temperature is 115°F. Extreme humidity conditions will also affect the crystal element, but when the crystal is properly coated, such microphones may be exposed to humidities of greater than 85% or less than 30% for as long as a month. Extended use under these conditions should be avoided.

Barium Titanate is a less sensitive piezo-electric crystal which is not affected by humidity. Such microphones have about the same frequency range as Rochelle salt microphones and have a dynamic range of 35-160 db. They are relatively unaffected by temperature in the range of 10-100°F and may be used in temperatures as high as 200°F provided the diaphragm or cements used in their construction do not soften or scorch.

Ammonium Dihydrogen Phosphate (ADP) is relatively expensive and of low sensitivity. It has a frequency range extending to 10,000 cps ± 2 db and up to 20,000 cps ± 4 db. Its dynamic range is 55-190 db. These microphones are little affected by temperature and may be used up to 250°F but should not be exposed to humidity of greater than 90% for over a month. A preamplifier must be used directly at the microphone.

Condenser Microphones - these have a frequency range extending to 10,000 cps and a dynamic range of 35-140 db. The temperature coefficient of sensitivity is 0.025 db per degree Fahrenheit rise. High humidity may cause noisy operation due to leakage across the condenser but dessication will restore quiet operation. A preamplifier must be used with this type of microphone.

Moving Coil or Dynamic Microphones. They are used where rapid fluctuations or extremes of temperatures or humidity must be tolerated. These microphones have a frequency response of 40-8000 cps $\pm$ 5 db and a dynamic range of 20-140 db. Because the nominal electrical impedance is about 20 ohms, relatively long connecting cables may be used without the necessity of applying temperature corrections. A relative humidity greater than 90% may adversely affect the cements used in the diaphragm. To connect a dynamic microphone directly to a sound level meter or amplifier, a matching transformer whose turns ratio is about 30:1 is necessary.

Because of microphone directionality, it is usual to calibrate a microphone for at least two types of sound fields. When the microphone is in a reverberant chamber, repeated reflections from the walls result in a diffuse sound field, with sound incident on the microphone at random angles. A different microphone response will be obtained for sound at grazing incidence. See Fig. 2.5 for representative response curves.

When using a microphone in an intense sound field (above 140 db) it is sometimes found that the frequency response of the meter is drastically changed. This is because such sound levels can set up vibrations in the microphone housing itself and these vibrations are transformed into an electrical signal along with the diaphragm movement.

A windscreen 8/ is necessary when a microphone is used in a wind or air stream. It consists of silk or muslin stretched on a cylindrical or spherical wire frame completely enclosing the microphone. In high winds it may be necessary to enclose the first windscreen by a second screen. The noise caused by turbulence around the microphone housing can be reduced by 30 db with such a screen.

Sound Level Meter

The basic instrument for measuring sound levels is the sound level meter. In order to allow the comparison of data taken by different investigators, it was found necessary to set up standards for sound level meters. See American Standards on Sound Level Meters Z 24.3 (1944), American Standards Association. Meters which meet these standard specifications are manufactured by General Electric Company, General Radio Company, H. H. Scott, Inc., Western Electric Co. and others. The instrument

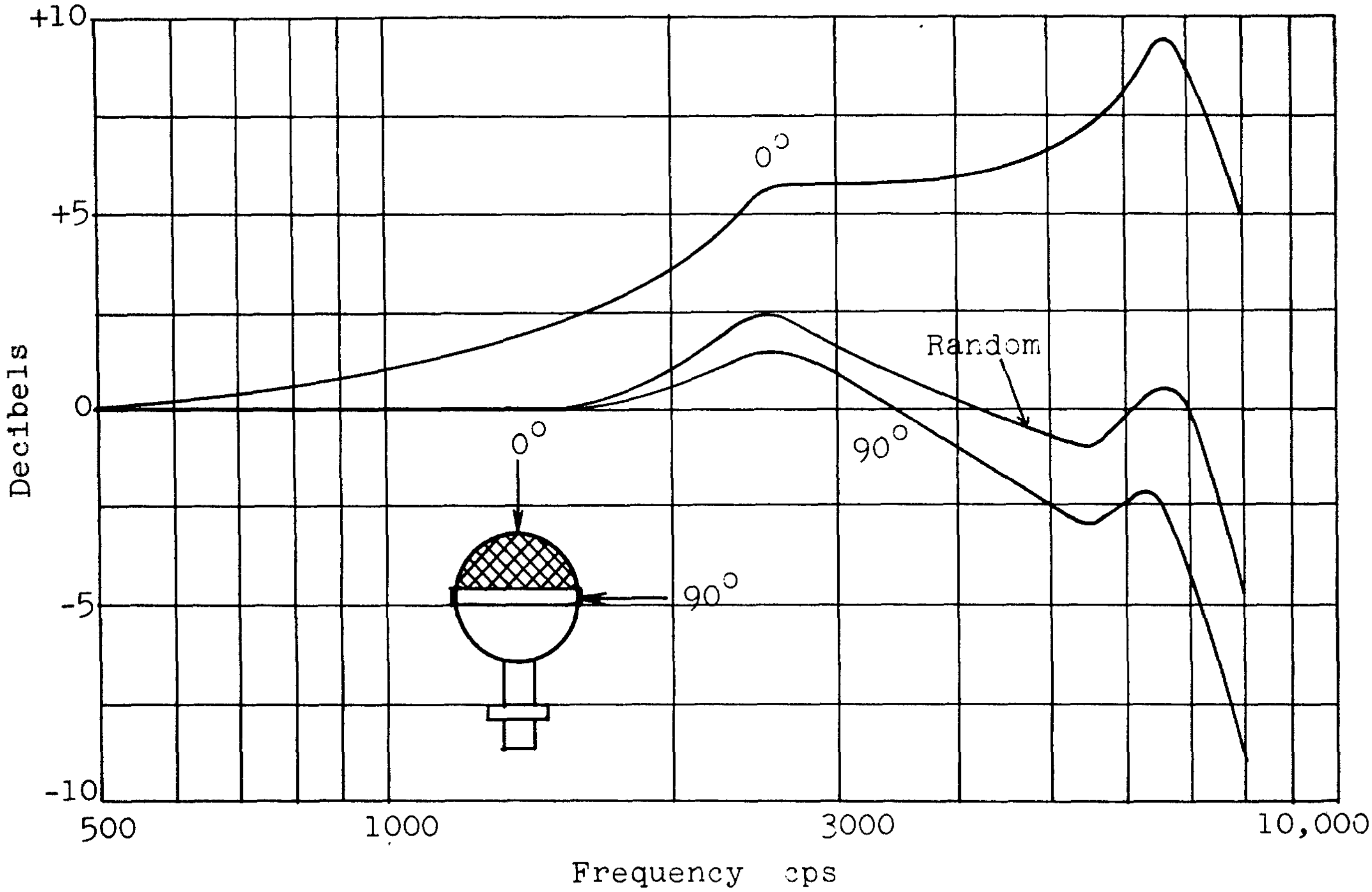


Figure 2.5

Field-free frequency response of a Rochelle salt crystal microphone for sound incident at 0°, 90°, and incident randomly (courtesy General Radio Company, Cambridge, Mass.).

contains an amplifier, a calibrated attenuator, and a switch which provides three different frequency responses. One of these responses is flat while the other two approximate the frequency response curve of the normal ear at two levels.

A sound level meter must be both portable and stable. The first of these requirements can be met by making it battery operated. For additional convenience some manufacturers have designed a power supply which can be substituted for batteries when operation near a source of power is possible. Stability of the meter's amplifying circuits and indication is obtained by use of negative feedback. A day-long stability of 0.5 db has been reported with the General Radio Type 1551A SLM. Regardless of even this high degree of stability a SLM should be

calibrated at least at the beginning and end of a day's measurement. Two factors which will limit the lowest sound pressure levels that can be measured are the internal noise level of the microphone and the noise level of the first stage of the SLM amplifier. While the noise level of a condenser microphone is equivalent to 35 db, Rochelle salt and dynamic microphones are so quiet that amplifier noise is the limiting factor. Using either of these microphones, the noise level of the SLM amplifier should permit measurements down to 25 db.

Some sounds will have peak values which are 5 to 15 db above average value. The SLM amplifier must have a dynamic range that can handle these peak levels without overloading. There are two further requirements on the SLM. Because the sound level meter may be used in a vehicle or near machinery, it should be unaffected by vibration. That is, the vibration should neither impair the functioning of the meter nor should it result in spurious meter readings. And secondly, the SLM should be sufficiently free from microphonics so that it can be used in high level noise fields. Acoustic vibrations may be transmitted directly through the meter case or support and cause the meter to read too high. If there is a possibility of this happening, the SLM should be connected to the microphone by a long cable, and removed as far from the noise source as possible. In even more extreme cases a barrier can be erected around the SLM, but in such an instance, the damage risk to hearing for the human operator may be too high to tolerate.

Filters and Noise Analyzers

While the overall SPL as read directly with a SLM may be an adequate index of acoustic conditions in some problems, it is usually desirable to obtain some knowledge of the relative amplitudes of the frequency components present in the sound. This is done by connecting the output signal of the SLM to some sort of electrical frequency analyzer. The simplest analyzer is the cathode ray oscilloscope. Peak values can be read directly and the fundamental frequency can, for some waveforms, be obtained from knowledge of the oscilloscope sweep rate. An experienced observer can often estimate what harmonics are present. A photograph of the waveform can be used to make a more elaborate harmonic analysis by means of a Henrici analyzer. [9,10]

Electrical filter sets are of two basic types: constant bandwidth analyzers and proportional analyzers. The former are rather expensive laboratory instruments which are not readily

portable. A common bandwidth is 5 cps although bandwidths of 200 cps are available. Narrow bandwidths impose stringent requirements on the sharpness of the bandwidth selector, and the use of these instruments is apt to be tedious when investigating a wide band of frequencies, say 20-10,000 cps. Also the fundamental frequency of the waveform being measured must be quite steady, or it and its harmonics would shift in and out of the narrow pass band during the course of the measurements. Proportional analyzers can be broken down into two categories: constant-percentage narrow-band and octave-band instruments. In the constant bandwidth instruments considered above a single frequency range Δf describes the instrument. For proportional analyzers $\Delta f/f_0$ is constant, where f_0 is the midband frequency. Thus for example, the bandwidth centered about 900 cycles is twice as large as the bandwidth centered around 450 cycles. As a consequence the number of measurements necessary to cover the usual audio band of frequencies is considerably reduced because at the high frequencies the pass band is proportionally widened. For an octave band analyzer, the upper cut-off frequency of each band is double the lower cut-off frequency. This corresponds to a value of $\Delta f/f_0$ of 2/3. A common set of octave bands are 20-75, 75-150, 150-300, 300-600, 600-1200, 1200-2400, 2400-4800, 4800-10,000 cps. It will be noted that the first and last bands are slightly wider than an octave in width. In almost all of the work to be described in this manual, the octave band analyzer (OBA) has been used.

If a continuous noise source containing equal amounts of energy in all frequency bands (see Fig. 2.6a) were measured with a constant bandwidth analyzer, the result shown in Fig. 2.6b would be obtained.

With an octave band analyzer, however, as each measuring band became wider, more sound energy would be recorded. Because of the doubling of bandwidth per octave, the meter output would increase at the rate of 3 db/octave, thus it would seem that the OBA measurements would give a false description of the noise source. The criteria to be presented and the characteristics of noise sources and sound treatments in the subsequent sections are given in most cases in octave band form, and hence measurements made with an OBA can be compared directly with these curves.

Recorders

It is often desirable to have scme form of a permanent record of a noise, either to study its time variation or to make more complex measurements than can easily be made in the

field. Magnetic tape recorders are often used for this purpose. If valid results are desired, the recorder and its associated playback system must have a flat frequency response, low hum and noise level, low distortion and a wide dynamic range.

Another useful instrument for such applications is the graphic level recorder. Here the position of a pen or stylus trace on a moving strip of paper is related to the instantaneous value of the quantity being measured. Writing speeds of 750 db/sec with a 50 db range on recording paper 5 cm wide are available.

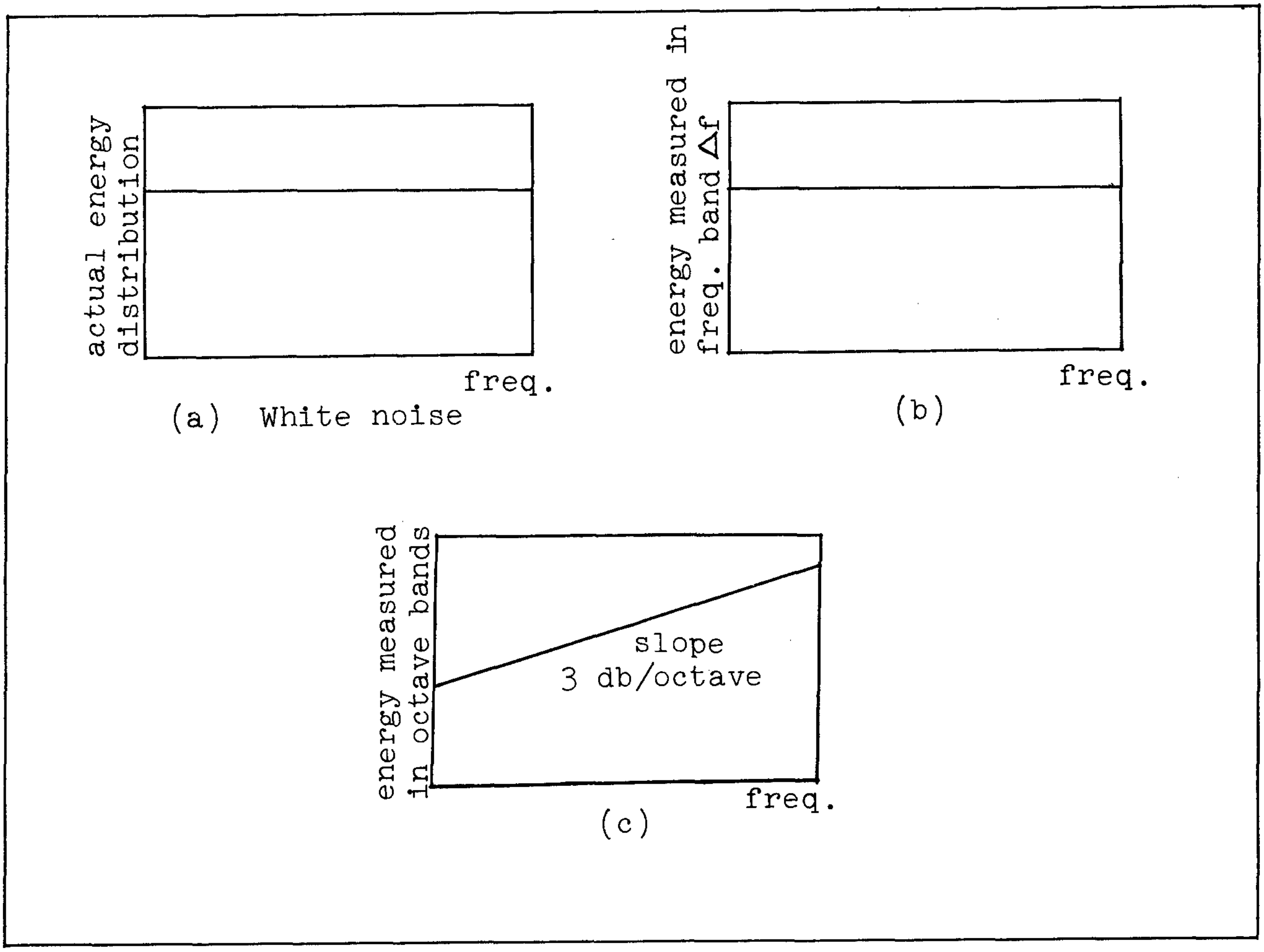


Figure 2.6

Apparent change in spectrum shape caused by the measuring instrument. Figure (a) gives the original noise spectrum; in this case it is "white" noise which has equal amounts of energy in all frequency intervals. Figure (b) shows the spectrum which could be obtained with a constant bandwidth measuring instrument. Figure (c) shows the result of measuring octave bands.

Calibrators

A calibrator is a device which will furnish a reproducible sound field to be used for the purpose of checking the measuring instruments. One model, made by the General Radio Company, is shown in Fig. 2.7. It consists of a battery operated 400 cps oscillator, a voltmeter and attenuator for adjusting the electrical output to a designated value, a stable loudspeaker, and provision for accurately positioning the microphone with respect to the speaker. The speaker housing shown is made for use with the Western Electric 633 A and a Shure Rochelle salt microphone used on some models of General Radio Company sound level meters.

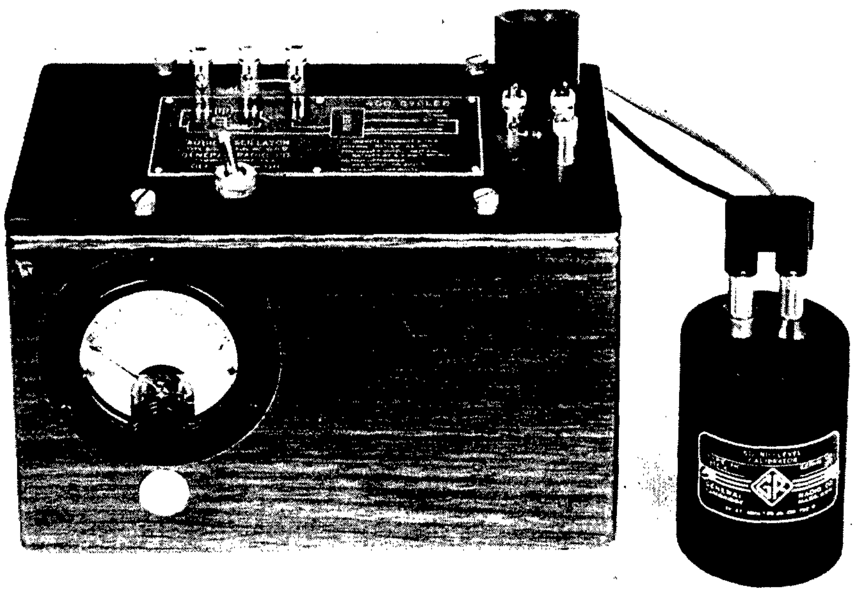


Figure 2.7

The General Radio Type 1552-A sound-level calibrator (Photograph courtesy General Radio Company, Cambridge, Mass.).

The calibrator is shown in use in Fig. 2.8. Here the overall sensitivity of the microphone and SLM is being checked. With a 2 volt signal applied to the speaker and with the calibrator in place over the microphone, the SLM should read 85 db. A second use of the calibrator is in determining the cable correction discussed in the next section under "data reduction". To evaluate the loss in the cable, the signal generated by the calibrator is measured with the microphone connected directly to the SLM (as in Fig. 2.7) and with the microphone connected to the SLM through the microphone cable. The decibel difference between these readings is the cable correction.

Figure 2.8

The sound-level calibrator being used to check the 400 cycle response of a microphone and sound level meter (Photograph courtesy General Radio Company, Cambridge, Mass.).

The equipment should be calibrated at the start and at the end of any series of measurements. If the ambient temperature varies over wide limits or the measurements require a considerable length of time it is advisable to check the calibration even more frequently. By taking such precautions, the measurements should be reliable to ± 0.5 db.

2.4 Measurements: Data Reduction

The goal of data reduction is to present the results of a set of measurements in the simplest possible form which will still contain all the necessary information. For example, if several measurements were made under the same conditions, some representative value should be obtained. Various corrections, such as will be discussed in detail below, should then be applied. The next section will treat more fully the methods which may be used to present the reduced data.

Data can be either of an absolute or a relative nature. In specifying the directionality of a sound source, it is usually sufficient to know how many db lower is the sound in any direction, relative to some reference direction. For such data it is not necessary to make any corrections for the microphone response, insertion loss of the microphone cable, etc., because one is merely measuring decibel differences. However, when the noise level of a jet engine is specified as say, 93 db (re .0002 dyne/cm^2), this is a statement of the rms sound pressure in absolute physical units. For such absolute values, the data as taken must be carefully corrected to allow for all interfering effects.

When data are being taken, extreme care should be exercised to insure that all relevant facts are recorded. These records should include: (1) a complete list of the types and serial numbers of equipment (particularly to note which microphones and cables are used), (2) ambient temperature and humidity, (3) the locations and method of testing, and (4) the personnel making the measurements. The first two pieces of information will be necessary if absolute levels are desired. The last two may be necessary in connection with interpretation and checking of the data.

In general, the absolute calibration (relation between rms sound pressure level and microphone output voltage) as a function of frequency must be known. Such an absolute calibration can be obtained by means of the reciprocity technique 11/ or by a comparison with another microphone whose absolute calibration is

known. In addition, the dependence of the microphone upon temperature, humidity, barometric pressure, type and length of cable used must be known. Whenever absolute sound levels are desired, there are two basic corrections which must be applied: a cable correction and a microphone correction. The cable correction is also a function of temperature and so in some cases the temperature change must be considered.

Cable Correction. The cable correction can be calculated if the capacitance of the microphone as a function of temperature is known. The correction in decibels to be added to all sound level meter readings is

$$[\text{Cable correction}] = 20 \log_{10} \left(1 + \frac{C_c}{C_m}\right) \qquad \text{db}$$

where C_c is the capacitance of the cable and C_m is the capacitance of the microphone.

However, a simpler procedure is to measure the same sound under the same conditions first with the microphone attached directly to the sound level meter and then with the cable connected between the microphone and sound level meter. Subtracting the two decibel values will give the attenuation introduced by the cable, in decibels. This need be done at one frequency only, since this correction is the same for all frequencies.

Temperature Correction. Since the dielectric constant of the piezo-electric elements of a crystal microphone is temperature dependent, the internal impedance of this type of microphone, which is essentially capacitative, is also dependent on the surrounding temperature. When the microphone is connected to a high impedance this variation in capacitance is of little effect. For example, the input resistance of the General Radio Type 759B sound level meter which normally employs a crystal microphone is 7 megohms. When a cable is used, the microphone is then connected to the high input impedance of the sound level meter shunted by the distributed capacitance of the cable. For a 25 ft length of cable this capacitance may be 600 $\mu\mu$ f. A typical microphone of this type may have a capacitance on the order of 2000 $\mu\mu$ f. Because the microphone capacitance varies with temperature, a correction must be made if the microphone is used at a temperature different from that at which the above cable correction was determined.

For example, suppose that at room temperature ($70^{\circ}F$) the cable correction is 2.5 db. That is, all SLM readings are too low by 2.5 db and 2.5 db must be added to the readings. If now some measurements are made near the exhaust stream of a jet engine when the temperature is say $100^{\circ}F$, an additional temperature correction is required. Suppose the microphone capacitance is $100^{\circ}F$ to be 800 $\mu\mu$ f and the cable capacitance to be 675 $\mu\mu$ f. From the expression given above, the total cable correction at $100^{\circ}F$ is 5.3 db, or 2.8 db more than the same correction at $70^{\circ}F$.

Microphone Correction. As pointed out previously, the absolute calibration of the microphone must be known is absolute sound pressure levels are desired. The relative response of the microphone must always be known, however, if variations in the microphone response are to be corrected for. In principle, each frequency should be corrected separately since the microphone response will vary with frequency. In a subsequent section the microphone correction for octave bands will be discussed. Because data are usually taken on some form of proportional bandwidth analyzer, the characteristics of these instruments are discussed more fully there.

Change in Cross-Sectional Area. A change in cross-sectional area, such as that shown in the schematic drawing (Fig. 2.9) will produce a difference in the sound pressure level at the point where the area change occurs according to the formula:

$$\text{Increase in Sound Pressure Level} = 10 \log_{10} (S_1/S_2) \quad \text{db.}$$

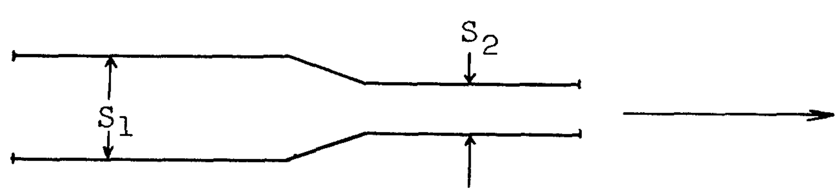


Figure 2.9

The change in sound pressure level due to a change in the cross-sectional area of a duct.

The assumptions made here are that the ambient pressure and the ambient temperature are not different on the two sides of the area boundary and also that there is no loss of energy across the area boundary.

Changes in Air Stream Velocity, Ambient Pressure, and Ambient Temperature. If a sound wave traverses a region in which there is a change in the ambient pressure, air stream velocity, ambient temperature, or any combination of these, the sound pressure level at each point is:

$$SPL = PWL - 10 \log_{10} S + 10 \log_{10} [P_o/(c + v)] + 15.5 \quad \text{db}$$

where

PWL = power level

S = cross-sectional area of the region in sq ft

P_o = ambient pressure in inches of mercury

c = speed of sound in feet per second

v = speed of air flow in feet per second (positive when in the direction of the sound propagation). This quantity is also called the windstream velocity.

If there is no dissipation of acoustic energy when the wave traverses the region of the changing condition, and if no change in cross-sectional area occurs, then:

$$SPL_1 - SPL_2 = 10 \log_{10} \frac{P_{01}\,(c_2 + v_2)}{P_{02}\,(c_1 + v_1)} \quad \text{db.}$$

In the case of heat exchange, the ambient pressure remains constant and the windstream velocity is proportional to the absolute temperature. The change in sound pressure level becomes:

$$SPL_1 - SPL_2 = 10 \log_{10} \frac{(c_2 + v_2)}{(c_1 + v_1)} \quad \text{db.}$$

If the velocities v_1 and v_2 are small compared to c_1 and c_2, respectively, then the sound pressure increases 1.5 db for each halving of the absolute temperature. If v_1 and v_2 are large compared to c_1 and c_2 then the sound pressure increases 3 db for each halving of the absolute temperature.

In many cases, c_1 and v_1 are about of the same magnitude while v_2 is small compared to c_2. In this case the change is more nearly 1.5 to 2.0 db than 3 db for each halving of the absolute temperature.

For reference, the volume of a given weight of air is given by

$$V = W \times (12.4) \times \frac{T}{273} \times \frac{30}{P}$$

where

V = volume of air in cubic feet

W = weight of air in pounds

T = absolute temperature in degrees Kelvin, and

P = absolute pressure in inches of mercury.

Noise Reduction Through the Piping Sidewalls. Some of the noise inside pipes will pass through the sidewalls and be radiated outside. In doing so, the sound pressure levels are reduced in magnitude. Because of the inherent difference in stiffness of circular pipes as compared to flat surfaces, one would expect the noise reduction through pipe sidewalls to show quantitative differences from estimates based on attenuation data for flat plates. This is examined in detail in Section 11.5.

Data Presentation. Experimental data are usually presented in the form of a table of values or a graph. Tabular presentation usually involves few difficulties. Each table should be numbered, and have an appropriate descriptive title, e.g., "Some Properties of Compressed Hydrogen at 0°C". Then for each column in the table three pieces of information should be given: the physical quantity listed, the mathematical symbol given it in the text, and the units. Under no circumstances should the units be omitted.

Graphs are useful for the visual interpretation of data. Qualitative graphs, used to show such things as the distribution of the tax dollar, the composition of the earth's crust, et., are familiar to everyone. They may consist of horizontal or vertical bars, segments of a circle, or little pictures of varying size or number. More important, however, are the quantitative graphs which present actual measurements and may be used in interpolating,

extrapolating, smoothing data, evaluating derivatives and integrals, or merely to indicate a trend or characteristic of the data. There are certain rules and conventions which should be followed in their preparation. 12/

The first step in plotting a graph is to select the most appropriate coordinate system. Rectangular coordinates are the most usual. To represent data involving an angle as a variable, polar coordinates may be more suitable. When one or both variables being plotted cover a range of more than one or two powers of ten, rectangular coordinates having a semi-logarithmic or logarithmic scale may be preferable. Such scales have the effect of spreading the data so as to give all the plotted points equal weight. In addition, functions which are logarithmic or exponential in nature plot as straight lines on a logarithmic scale. For example, in the relation $y = ae^{bx}$, a and b are determined from the slope and y intercept of the straight line $\ln y = \ln a + bx$ obtained in a semilog plot of x and y. Here

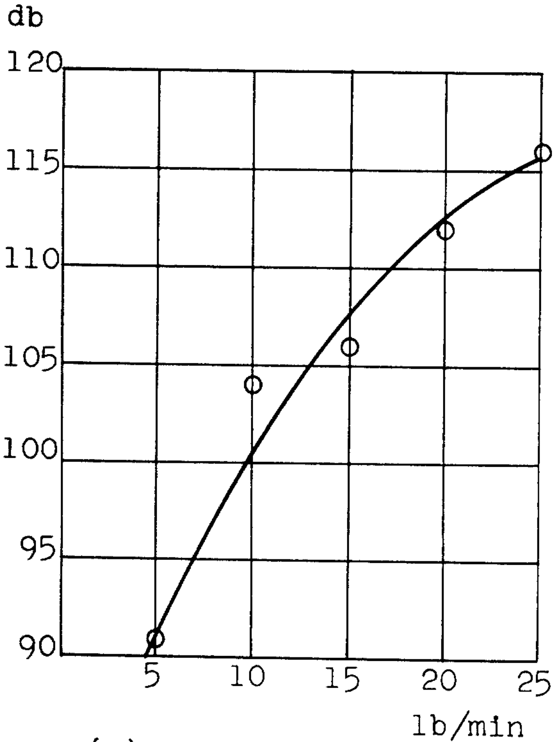
$$\text{slope} = b$$

$$y \text{ intercept} = \ln a.$$

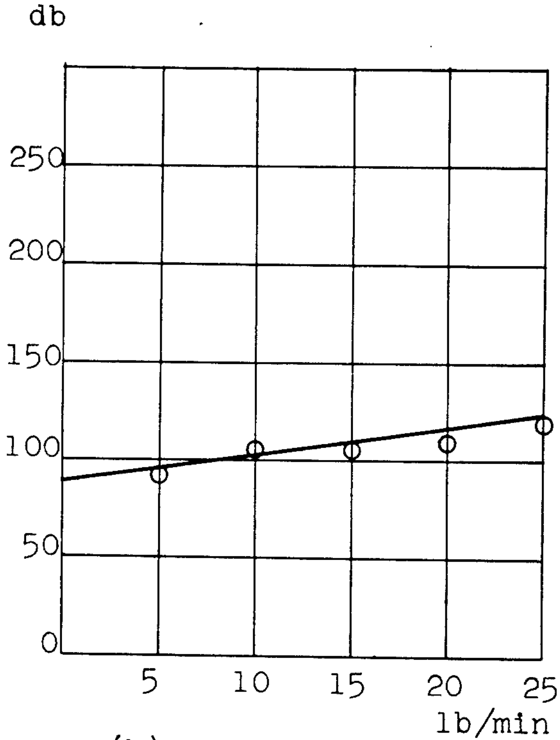
In addition, there exist special scales which are available, such as probability, etc.

Next the coordinate scales should be chosen. By convention the experimentally independent quantity is plotted as the abscissa, while the dependent variable is plotted as the ordinate. The scales should be chosen in such a manner that the coordinate of any point can be readily determined. In addition, a scale choice which aids in interpolation will make the plotting of the data easier. The extreme scale values should be chosen so as to cause the range of experimental data to extend over a majority of the graph, i.e., one or both scales should not be so compressed that all the data fall in one small area or band on the paper. On the other hand, the scales should not be so expanded that the data are made to look worse than is actually the case. A good rule to follow is to choose the coordinate scales so that the last experimentally determined significant figure can be plotted by interpolation between two of the smallest ruled divisions. The examples shown in Fig. 2.10 should make this point clear.

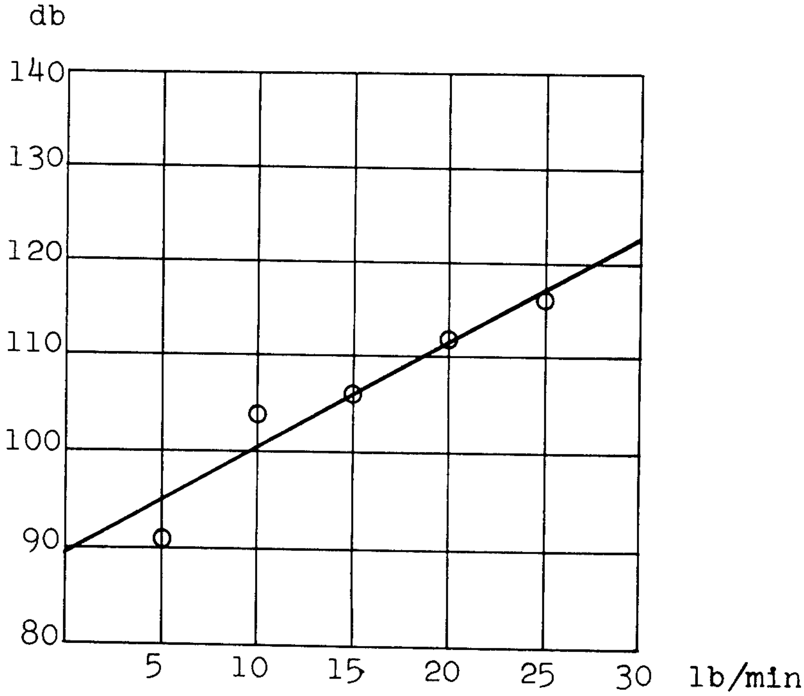
It is usual when plotting experimental data to indicate the probable error in value of the point being plotted. In most cases it is assumed that the independent variable is



(a) Coordinate scale overexpanded



(b) Coordinate scale overcompressed



(c) Correct choice of scales

free of error. Then the dependent variable is plotted as a point lying on a straight vertical line, the length of which measures the error or range of the measurements. Such a scheme is shown in Fig. 2.11. Suppose fifty measurements are taken ranging from 112 to 138, with a mean of 122 and a probable error of 5. Such a measurement could be entered on a graph in the forms shown either in Fig. 2.11a or Fig. 2.11b.

When measured values have been plotted on graph paper, the next step is usually to connect these points with some sort of continuous line or curve. There are two ways in which this may be done. Either a smooth average curve passing somewhere near all the points but not necessarily through any of them, may be drawn or successive points may be connected by straight lines. In this handbook both procedures are employed.

If the data are of such a nature that one would expect a well-behaved functional relationship, then it is permissible to draw a smooth curve. For example, even widely scattered values for sound absorption as a function of frequency for moist air as shown in Fig. 2.12 could be approximated by a smooth curve since a smooth regular behavior with increasing frequency is to be expected.

If there are no physical reasons why the data should be functionally related, or if there is doubt as to the behavior of the data between the measured points, the better procedure in plotting the information is to join the points by straight line segments. Thus a plot of instantaneous noise level of a certain aircraft engine taken at 10 second intervals might be as shown in Fig. 2.13. There is no reason to believe that the noise level should be a function of time, as indeed such data would indicate. Except for the random influence of a great many variables, the noise level is approximately constant. In this case the chief function of the straight line sections is to direct attention to the measured points and indicate the range of variation and the trend of the data.

Figure 2.10

Factors involved in the choice of scale when plotting data. Figure (a) shows the result of expanding the vertical scale beyond the accuracy of the experimental data. Figure (b) shows the result of compressing the scale, thus hiding any significant variation in the data. Figure (c) shows the correct choice of scale.

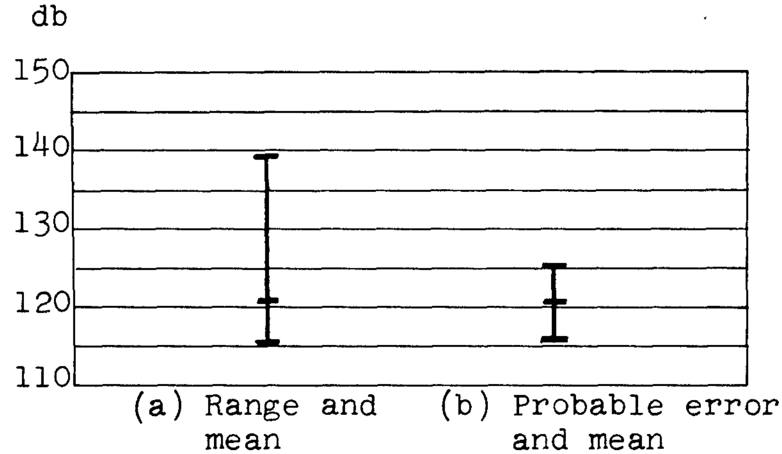


Figure 2.11

Methods of plotting experimental values. In (A) the mean is plotted and a vertical bar indicates the range of the measurements which make up the mean. Figure (B) shows the mean and a symmetrical vertical bar indicates the standard deviation or the probable error of the mean.

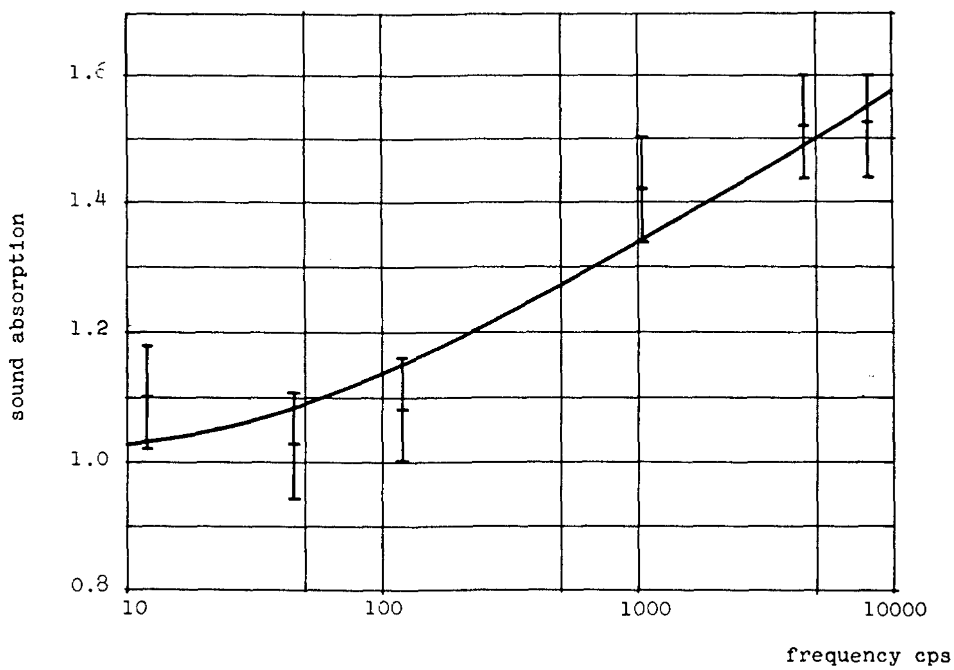


Figure 2.12

Hypothetical curve showing the use of a smooth curve to represent experimental data.

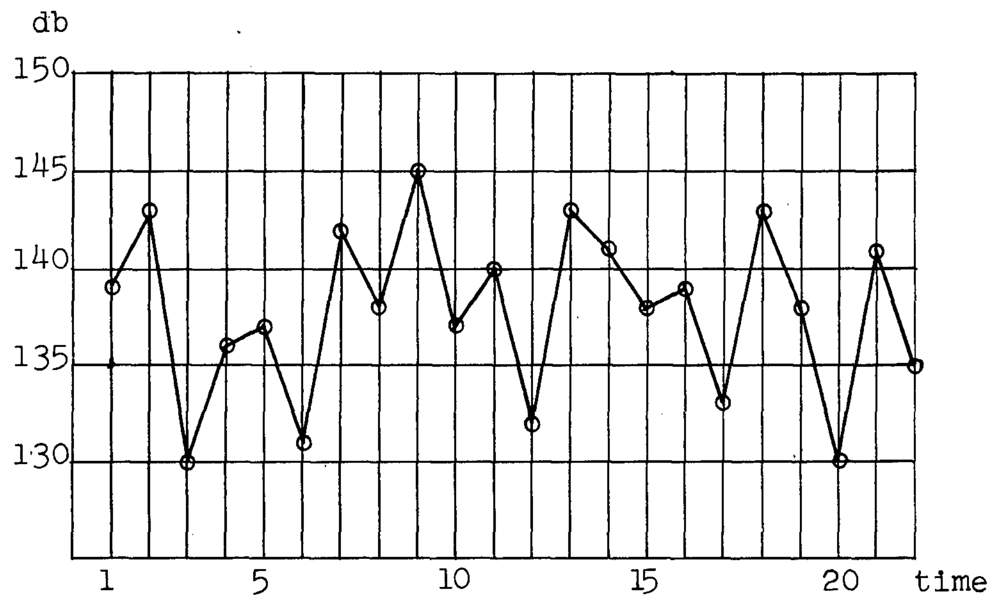


Figure 2.13

Hypothetical case illustrating the use of a broken line to represent experimental data.

Specific Acoustical Examples

1. Averaging decibel values

The two most common acoustic quantities which are expressed in decibels are acoustic power and rms sound pressure level. Since many instruments read directly in decibels, often all the data which are taken will be expressed in decibels. As discussed in previous sections, it is usually desirable to take several readings and find a single, average, representative value. There are methods for obtaining this average value: one may express each power or SPL in watts or dyne/cm^2, average those quantities, and present the result in db, or, one may average the decibel values directly. In general the two results will be different.

The first method consists essentially of finding the arithmetic mean of the individual powers or SPL and is the best method. It is a rather laborious process and if there is a large amount of data, may be quite time-consuming. In connection with the averaging of decibel values consider three powers, P_1, P_2, and P_3 (in watts). Expressed in decibels they become x_1, x_2, x_3. That is

$$x_1 = 10 \log_{10} \frac{P_1}{P_{ref}} .$$

Suppose that the geometric mean of the powers is calculated. By definition

$$P = \sqrt[3]{P_1 \ P_2 \ P_3}$$

Divide both sides of this equation by P_{ref} and take logarithms. Thus

$$\frac{P}{P_{ref}} = \sqrt[3]{\frac{P_1}{P_{ref}} \times \frac{P_2}{P_{ref}} \times \frac{P_3}{P_{ref}}}$$

$$\log \frac{P}{P_{ref}} = \frac{1}{3} \left(\log \frac{P_1}{P_{ref}} + \log \frac{P_2}{P_{ref}} + \log \frac{P_3}{P_{ref}}\right) .$$

Multiplying each side of this equation by 10, there results

$$10 \log \frac{P}{P_{ref}} = \frac{1}{3} \left(10 \log \frac{P_1}{P_{ref}} + 10 \log \frac{P_2}{P_{ref}} + 10 \log \frac{P_3}{P_{ref}}\right).$$

But each of these expressions is simply the power expressed in db. That is

$$x = \frac{1}{3} \ (x_1 + x_2 + x_3) \ .$$

Hence it is seen that the power P which corresponds to the arithmetic mean x of the decibel values x_1, x_2, x_3 is simply the geometric mean of the powers P_1, P_2, P_3. If the three values of power are given in db, it is much easier to average the db directly and obtain what is really the geometric mean of the powers, expressed in db.

The question remains, which is the more meaningful average, the arithmetic mean of the powers, or the geometric mean. In

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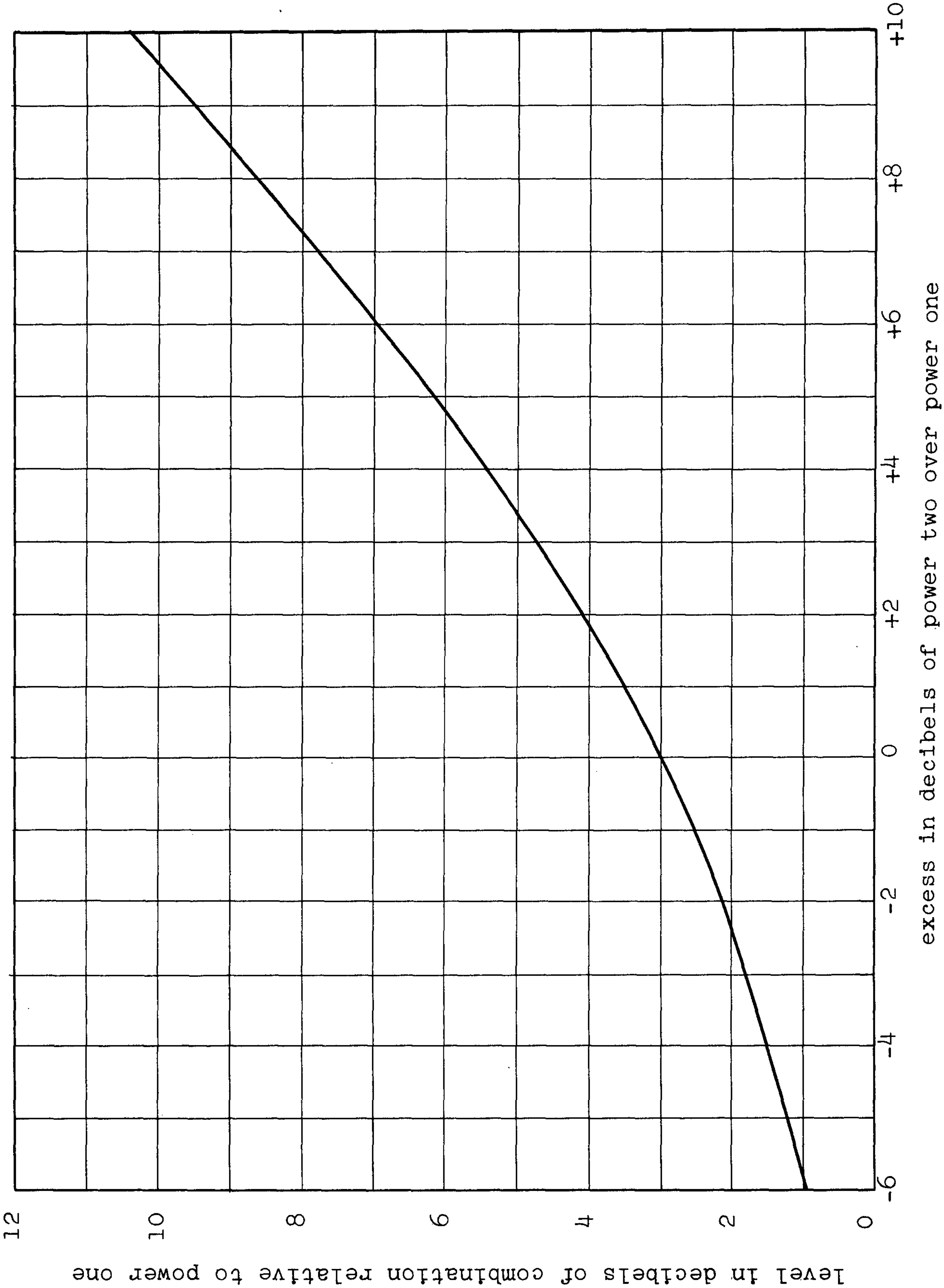
Figure 2.14

Chart for converting two powers, each of which is given in decibels, to their sum in decibels. To use, locate the difference in decibels between the two values on the abscissa and read the number of decibels which when added to the smaller power will give the sum of the powers, in decibels.


excess in decibels of power two over power one
-6
-4
-2
0
+2
+4
+6
+8
+10
level in decibels of combination relative to power one
0
2
4
6
8
10
12

general the arithmetic mean of the powers, obtained by converting all the powers from db to absolute physical units and averaging those values is more meaningful. Consider the following hypothetical case: a highly directional sound source radiates almost all of its power into one hemisphere and all over this hemisphere the SPL is 100 db; over the other hemisphere the SPL is a uniform 20 db. It is certainly misleading to say that the average SPL is 1/2 (100 + 20) = 60 db, since this would say that the average SPL is (re .0002 dyne/cm^2. If all the acoustic power were distributed uniformly over the sphere, the SPL would be 10 dyne/cm^2, corresponding to 94 db. This uniform distribution is what is intuitively meant by the word "average". Furthermore, 94 db indicates a high level sound field; while a 60 db average value would be unlikely to prepare one to find 100 db levels present in some places.

The case considered here was an extreme one, with an 80 db spread in the values being averaged. As the spread in values becomes smaller, the direct average of the db values becomes more meaningful. The value obtained by averaging decibels is always lower than the average obtained by converting to watts or dyne/cm^2. Therefore the following two rules can be stated:

(1) When the spread of decibel values is on the order of 10 db, average the db values and add 1 db. Depending on the distribution of values, the result will not be in error by more than ± 1 db.

(2) When the spread of decibel values is about 5 db, average the db values. This average will never be in error by more than 1 db.

Consider the following example. Eight experimental values are given as o, -1, -2, -3, -7, -8, -9, -10 db. Averaging these directly, one obtains -5 db. Adding 1 db gives an average of -4 db. It will be found on calculation that this value differs by only 0.6 db from that which would be obtained by the longer procedure of converting to absolute SPL. Since this error is within the usual range of accuracy of acoustic measurements, the simple db average is acceptable.

The chart shown in Fig. 2.14 will be found useful if one wishes to shorten the amount of calculation necessary to perform the arithmetic addition of two powers. Take the difference in db values between them and entering the chart with this value read on the abscissa the number of db to be added to the smaller db value. As an example, suppose one wishes to add two equal

powers of 86 db each. Since these are the same, adding doubles the power. From the chart, at zero db difference one adds 3 db to the smaller, giving 89 db. This is what would be expected since a doubling of power corresponds to a 3 db increase ($10 \log_{10} 2 = 10 \times 0.3 = 3$ db).

2. Octave-band Analysis of a Noise Field

One of the most valuable instruments for sound spectrum measurements is the octave-band analyzer described previously. The information provided by a closely-spaced set of measurements at individual frequencies is sacrificed to speed and convenience when it is used. The results of such an octave band survey are shown throughout the handbook, e.g., Fig. 7.7. It may be desirable in some instances to convert the octave band data to a continuous frequency scale. Assuming that the noise has a continuous spectrum and does not have any discrete frequency components, one subtracts from the octave band measurements $10 \log_{10} \Delta f$ where Δf is the width of that band in cycles. This value is then plotted at the frequency of the geometric mean of the band limits. Table 2.4 shows the results of the octave band measurements of Fig. 7.7 converted to a frequency scale.

TABLE 2.4

CONVERSION OF OCTAVE-BAND DATA TO A FREQUENCY SCALE

Octave Band	Measured power level in octave bands	$10 \log \Delta f$	Geometric mean of band limit	spectrum level
20-75	95 db	17	39	78
75-150	100	19	106	81
150-300	97	22	212	75
300-600	93	25	424	68
600-1200	88	28	849	60
1200-2400	90	31	1698	59
2400-4800	86	34	3390	52
4800-10000	85	37	6930	48

The quantity $10 \log \Delta f$ which is subtracted is the conversion from a bandwidth of one octave to one of one cycle. There is obviously less power in a one cycle band than in a band of say

1200 cycles, assuming a uniform continuous spectrum in that band. It is to be emphasized that this is an approximation which is valid only in the absence of any large variation in the spectrum. Since this cannot be assured, there is always an element of estimation involved in the above procedure.

The importance of narrow-band analysis is illustrated by the following example. Assume that an acoustic treatment has an actual attenuation vs. frequency characteristic as shown by curve A of Fig. 2.15, and that measurements are made using a wide-band noise source. A narrow-band measurement at the input and output of the treatment would yield curve A. If, however, we take measurements in wide bands (for example, octave bands) as indicated by the regions 1, 2, and 3, and plot the measured attenuations at the geometrical mean frequencies of the bands, we would obtain curve B which would give an erroneous evaluation.

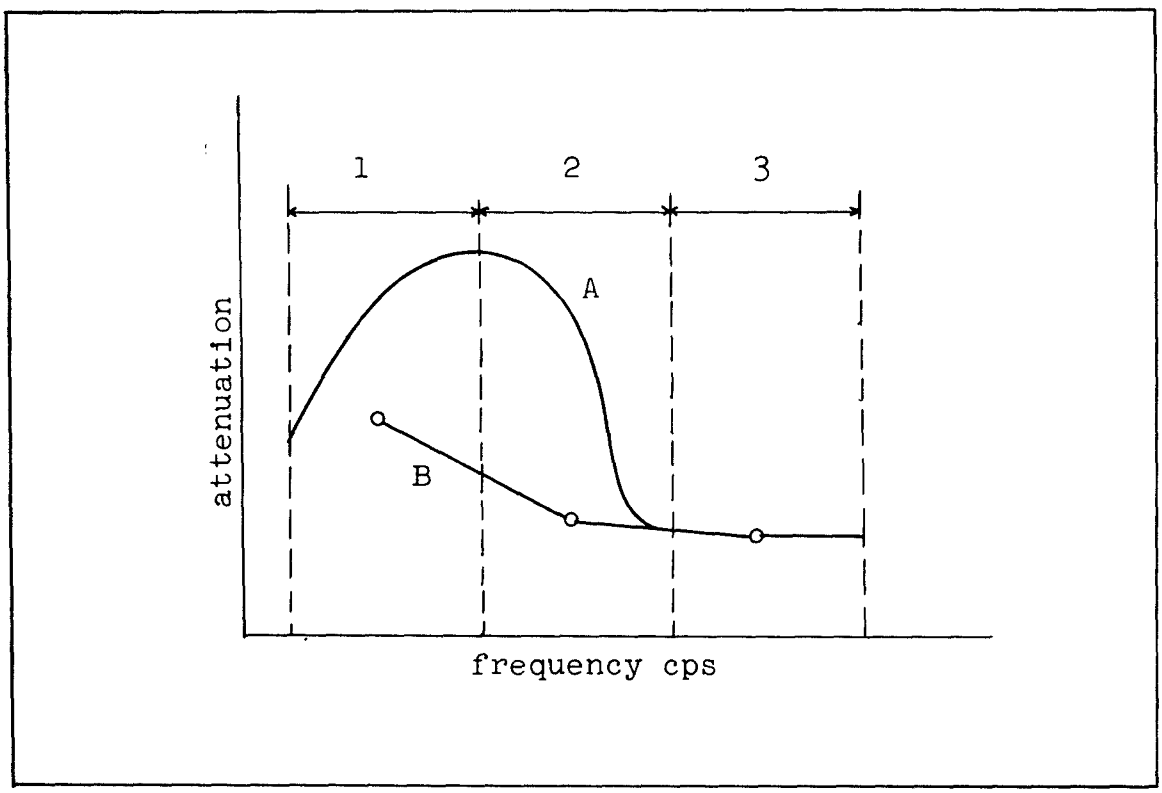


Figure 2.15

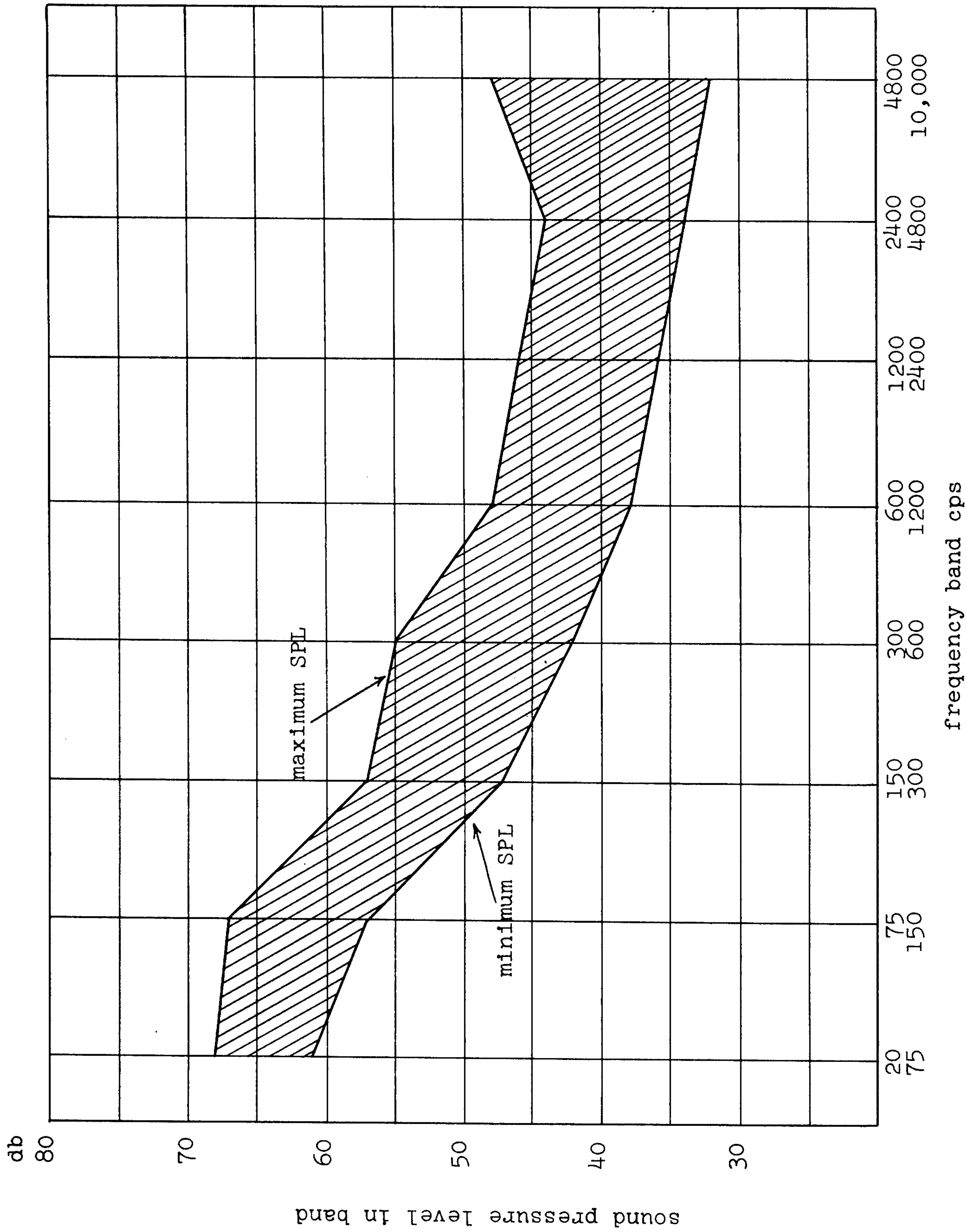
Comparison of wide-band and narrow-band frequency measurements of a sound treatment attenuation.

These differences occur because the octave band analyzer measures the total energy, without regard for frequency within a band. If the level varies considerably within the limits of a band, most of the energy is contained in those frequencies having the highest sound levels, i.e., in the regions of lowest attenuation within the band. Therefore the attenuation measured by octave bands will correspond approximately to the lowest actual attenuation within each band. If the attenuation vs. frequency curve is flat (as shown in band No. 3 of Fig. 2.15), the two curves will coincide. If a noise source which does not have a flat frequency spectrum (such as an engine) is used, narrow-band analysis becomes even more important for determining attenuation characteristics correctly. Since the octave band analyzer essentially measures the highest SPL present in an octave band, it gives a conservative, maximum figure in such cases for the amount of acoustic power present in the band.

3. How to Read a Sound Level Meter

By its nature, noise is a random phenomenon which exhibits a great range of instantaneous amplitude and frequency variation. This source variation, combined with short-term environmental changes which affect sound propagation tend to produce a continuous time variation in the sound-indicating instrument. This time variation can be smoothed considerably by incorporating into the measuring instrument a logarithmic response, so that the sound level meter indicates sound level directly in decibels. But even in this case the indication on such an instrument will fluctuate over a range of as much as 15-20 db. For this reason instrument designers incorporate a suitable damping resistance across the meter movement. There is usually a switch which enables one to insert or remove this extra resistor as necessary.

The actual technique of reading a sound level meter depends on the type of information which is desired. A few examples will make this clear. Suppose that one is measuring the background noise in a residential neighborhood. A probable variation of 20 db or more may be expected. Passing trucks, cars, and airplanes will cause unusually high levels, while there will be quiet moments when only such noises as the rustling of the trees will be heard. There is no real "average" sound level. In such a case a meaningful measurement can best be made by recording the maximum and minimum levels in each octave band, occurring in say a 30 sec period. A plot such as shown in Fig. 2.16 will result.



db
80
70
60
50
40
30
sound pressure level in band
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps
maximum SPL
minimum SPL

As a second example, consider the measurement of noise inside a factory building. There is a general ambient noise level which is occasionally raised due to some loud but relatively infrequent operation. In such a case the SLM indication would vary about some average value and occasionally read off scale. There the best thing would be to observe the instantaneous indications for several minutes, ignore the occasional peaks, and try to average by eye the mean reading. This means that there is a time-weighting involved in which the observer tries to estimate the "time-average sound pressure level". According to the discussion on averaging db, such an averaging process introduces an inherent error and is less meaningful as the range of variation of the observed noise level increases.

In the above example, the SLM indications cannot be read to closer than one db. However, when making measurements under controlled conditions, with a pure tone source, amplifiers, etc., then measurements to a fraction of a db become possible. The uncertainty in the exact temperature and cable correction usually makes such precise measurements doubtful in the last significant figure. One should not try to specify a sound level of 88.3 for example, when a variation of 15 db was observed while making the measurements.

2.5 The Literature of Acoustics

The following material will be found useful. Many of the references in this manual are to sources listed below. As a further guide to the reader, comments have been added in an effort to give some indication of the type of material covered in each case.

Theory

Bolt, R. H. and Morse, P. M., "Sound Waves in Rooms" Revs. Modern Phys. <u>16</u> 2 (1944).

A theoretical treatment of the interaction of a sound field and the room as a function of the room shape and the material lining the inner surfaces.

Figure 2.16

Background noise in a residential area. The SPL in each octave band was observed for 30 sec and the maximum and minimum values recorded.

Lamb, H., Dynamical Theory of Sound (Arnold and Co., London, 1931).

Morse, P. M., Vibration and Sound (Mc Graw-Hill Book Co., Inc., New York, 1952).

A mathematical treatment of vibration and sound, of an introductory nature.

Rayleigh, J. W., The Theory of Sound (Dover Publications, New York, 1945).

A recent reprint of an early classic work in the field of acoustics.

General Texts

Kinsler, L. E. and Frey, A. R., Fundamentals of Acoustics (J. Wiley and Sons Inc., New York, 1950).

A good introductory text which stresses the physical rather than mathematical side of the subject.

Knudsen, V. O. and Harris, C. M., Acoustic Designing in Architecture (J. Wiley and Sons Inc., New York, 1950).

This text covers those phases of applied acoustics as they concern architectural planning and design.

Olson, H. F., Elements of Acoustical Engineering (D. Van Nostrand Co., Inc., New York, 1947).

Acoustics as it is related to practical engineering and the design of electro-acoustic instruments.

Olson, H. F., Dynamical Analogies (D. Van Nostrand Co., Inc., New York, 1943).

A discussion of analogous electrical circuits which are frequently used to study acoustical systems.

Sabine, H. F., Less Noise, Better Hearing

An introduction to architectural acoustics, prepared by the Celotex Corporation, Chicago, Illinois.

Measurements

Beranek, L. L., Acoustic Measurements (J. Wiley and Sons, Inc. New York, 1949).

A comprehensive survey of equipment and measuring techniques.

Beranek, L. L., Apparatus for Noise Measurement (1950).

A discussion of noise measuring equipment, prepared for the General Radio Company, Cambridge, Massachusetts.

The catalogues and technical literature supplied by manufacturers of acoustical equipment is a useful source of detailed information on the performance characteristics of sound level meters, sound analyzers, etc. The publications of the following companies are suggested:

General Radio Co., Cambridge, Mass.

Western Electric Co., Inc., New York, New York

H. H. Scott, Cambridge, Mass.

Miscellaneous

Sabine, W. C., Collected Papers on Acoustics (Harvard University Press, Cambridge, 1927).

The original classic papers on architectural acoustics by one of the founders of the field.

Proceedings of the National Noise Abatement Symposium 20 Oct. 1950 (Armour Research Foundation, Chicago, Illinois).

Proceedings of the Second Annual National Noise Abatement Symposium 5 Oct. 1951 (Armour Research Foundation, Chicago, Illinois).

These two volumes above are composed of papers which were presented at the Noise Abatement Symposia of 1950 and 1951.

Journal of the Acoustical Society of America, Wallace Waterfall, Secretary, Acoustical Society of America, 57 E. 55th St., New York 22, N. Y.

This bi-monthly journal contains the latest work in all phases of acoustics.

Official Bulletin of the Acoustical Materials Association

This publication presents data on the current acoustical products of its members. Acoustical Materials Association, 57 E. 55th St., New York 22, N. Y.

American Standards Association, 70 E. 45th St., New York 17, N.Y.

Proposed or accepted standards on terminology, measurement procedures, etc.

References

(1) Hoel, P. G., Introduction to Mathematical Statistics J. Wiley and Sons, 1947.

(2) Mosteller, F., "On Some Useful Inefficient Statistics", Annals of Mathematical Statistics 17 377 (1946).

(3) Dixon, W. J. and Massey, F. J. Jr., Introduction to Statistical Analysis, McGraw-Hill, 1951, Ch. 15 - 17.

(4) Snedecor, G. W., Statistical Methods, Iowa State College Press, 1946, 4th ed., p. 98.

(5) Patnaik, P. B., "Mean Range as an Estimator of Variance", Biometrika 37 78 (1950).

(6) Beranek, L. L., "Apparatus for Noise Measurement", General Radio Company, Cambridge, Mass. (1951).

(7) Beranek, L. L., Acoustic Measurements, J. Wiley and Sons (1949) Ch. 5.

(8) Ref. (7) p. 258-9.

(9) Henrici, O., Phil. Mag. 38 110-125 (1894).

(10) Ref. (7) p. 580-586.

(11) Ref. (7) p. 113 ff.

(12) Worthing, A. G. and Geffner, J., Treatment of Experimental Data, J. Wiley and Sons, 1943.

PART II. NOISE SOURCE CHARACTERISTICS

CHAPTER 3

THE SPECIFICATION OF A NOISE SOURCE

3.1 Introduction

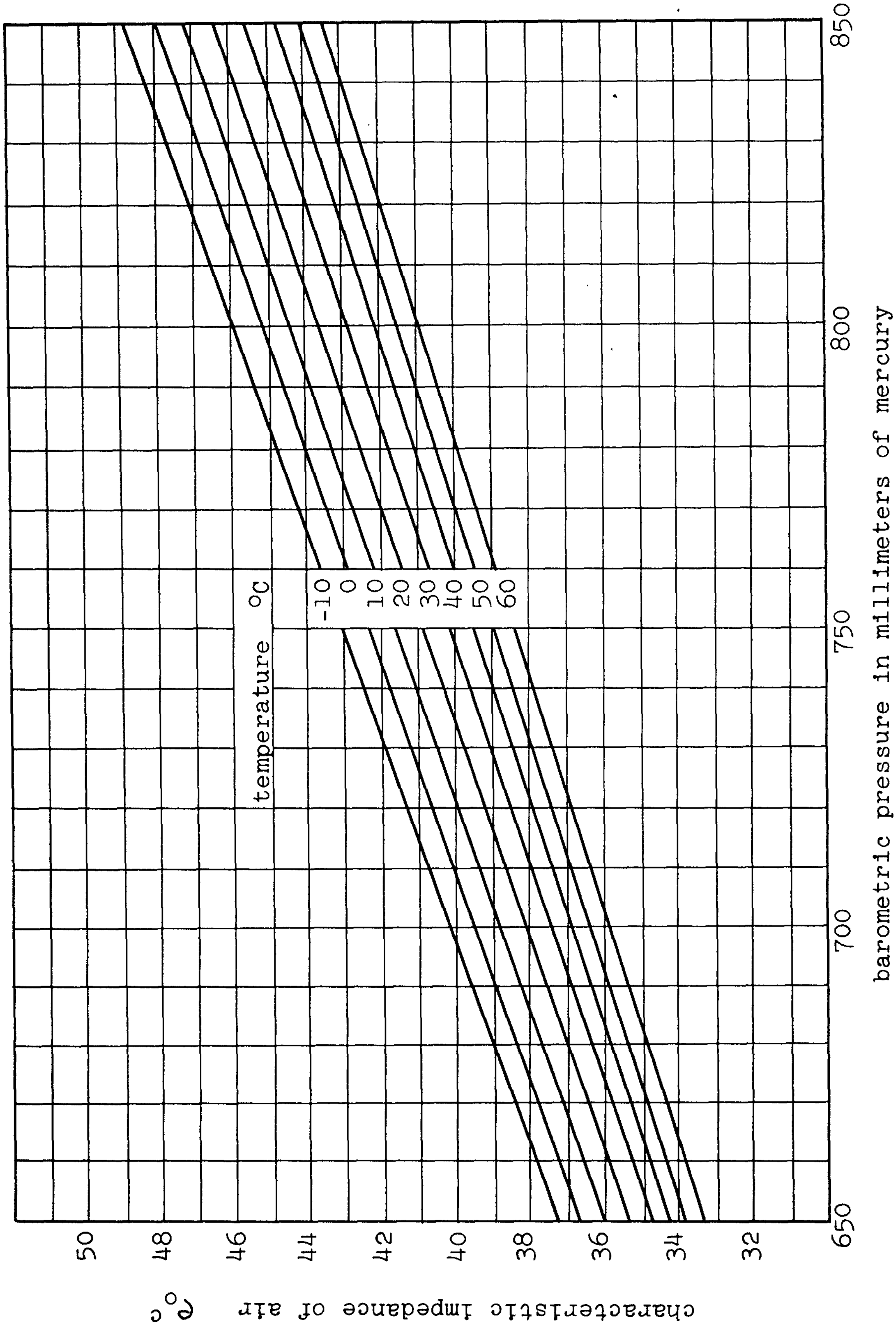
All vibrating objects will radiate sound. The character and amount of the radiated sound will depend upon the type of source and the properties of the medium into which the sound is radiated. In practice, the parameters that are needed to describe the source are (1) the total power radiated by the source, (2) the power radiated by the source as a function of frequency or frequency bands, (3) the directionality of the source as a function of frequency or frequency bands. The characteristics of the media into which sound can be propagated are so numerous and varied that the discussion will be restricted to those situations that are commonly encountered in analyzing noise sources. These are (1) free-field non-directional, (2) free-field directional, and (3) diffuse conditions. In the following paragraphs these terms will be discussed.

3.2 Free-Field Conditions, Non-Directional Source

The term free field is applied to an isotropic, homogeneous sound field free from bounding surfaces. In practice, a measurement of sound pressure level is often said to be made under free-field conditions when the following restrictions are realized: (1) No reflecting surfaces are present, (2) The sound pressure is not significantly influenced by the "near field" of the source, (3) The source is located in a homogeneous, isotropic medium.

In general, so-called near-field effects are avoided when all measurement points are separated from the source by at least several times the largest dimension of the source. Beyond this near-field region the radial variation in intensity is in accordance with the inverse square law.

Under free-field conditions, uniform spherical waves are produced either by a point source of sound or by a spherical source, of finite size, all points of whose surface vibrate


temperature °C
-10
0
10
20
30
40
50
60
650
700
750
800
850
barometric pressure in millimeters of mercury
50
48
46
44
42
40
38
36
34
32
characteristic impedance of air $\rho_0 c$

in phase. When uniform spherical waves are radiated, the sound pressure level is the same at all points that are equidistant from the source. In addition, the sound intensity satisfies the inverse square law, in which both the intensity level and the sound pressure level decrease 6 db for each doubling of distance from the source.

As far as the SPL measurements are concerned, there is no near-field disturbance in the case of a truly spherical source. The inverse square law holds for all points beyond the surface of the source. Certain practical sources of sound, which are not actually uniform spherical radiators, produce approximately uniform spherical waves at a distance, but are characterized by near-field irregularities.

When free-field non-directional radiation exists, a single measurement of the sound pressure level for a given frequency at a known distance from the source will be sufficient to determine the sound power radiated by the source. Thus,

$$SPL = 10 \log W + 10 \log \rho c - 20 \log r + 103.3 \text{ db} \tag{3.1}$$

where

SPL = sound pressure level
W = power of source in watts
r = distance from source in ft
ρc = characteristic impedance of the air in rayls (specific acoustic ohm)
$10 \log \rho c$ = 16.2 db for a temperature of 70°F and a barometric pressure of 30 in. (762 mm) of mercury

A chart showing the variations of ρc with temperature and pressure is presented in Fig. 3.1.

Power Level. Often the acoustic power of a source is expressed in terms of the power level (PWL), defined in Eq. (3.2)

$$PWL = 10 \log (W/P_{ref}) \tag{3.2}$$

Figure 3.1

The characteristic impedance of ρc of air, in rayls, as a function of barometric pressure and temperature. From Acoustic Measurements by Beranek.

It is desirable to choose a reference level which will be convenient for numerical calculations, and which will bring about a simple relation between SPL and PWL when the acoustic power of sound in air is uniformly distributed over a standard area of one sq ft.

The value of 1.00×10^{-13} watts will be used as the standard power reference level in this manual. When this value is used, the relation between SPL and PWL is found from Eq. (3.1) to be

$$\mathrm{SPL} = \mathrm{PWL} - 10 \log S + [10 \log \rho c - 15.7] \tag{3.3}$$

where S is the area in sq ft over which sound power is uniformly distributed. At 70°F, barometric pressure of 30 in. mercury, the term in brackets is 0.5 db. The uncertainty in many noise measurements is large enough to justify neglecting this correction. Therefore, to a good engineering approximation, the SPL and the PWL are numerically equal when the total power is uniformly distributed over an area of one sq ft. The correction term in brackets is zero when ρc has the value 37.2 rayls. This condition is realized at a temperature of approximately 196°F, when the barometric pressure is 30 in. mercury.

Other values which have been used for the reference power are 0.90×10^{-13} watts and 0.93×10^{-13} watts. When the value 0.90×10^{-13} is used, the relation corresponding to Eq. (3.3) includes a correction term which becomes zero for ρc equal to 41.3 rayls. When the value 0.93×10^{-13} watts is used the corresponding correction term becomes zero for ρc equal to 40.0 rayls.

The disparity between the extreme reference values of power (0.90×10^{-13} and 1.00×10^{-13} watts) is less than 0.5 db. Therefore, the experimental values for PWL of various noise sources, as given in this manual, are the same within the accuracy of the measurements when any of the power reference values given above are used.

Another convenient way of expressing the PWL in terms of the reference level of 1.00×10^{-13} watts is

$$\mathrm{PWL} = 10 \log W + 130 \quad \mathrm{db} \tag{3.4}$$

For a non-directional source in a free field, it is sometimes more convenient to express the relation of SPL

and PWL in terms of the distance r from the source, rather than in terms of the area covered by the radiation. Thus, if S in Eq. (3.3) is replaced by $4\pi r^2$, the equation becomes

$$SPL = PWL - 20 \log r + 10 \log \rho c - 26.7 \qquad (3.5)$$

3.3 Free-Field Conditions, Directional Source

Again consider a source that is situated in a region free of all bounding surfaces. This source is not radiating uniformly in all directions, either because of its shape, or because the vibrations of the elements making up the source are not in phase or not uniform or both. It will be assumed that measurements of SPL are made at a sufficient distance from the source to avoid near-field effects.

The properties of the directional sound will be specified in terms of the directivity factor, Q(f). The directivity factor is, at a given frequency, the ratio of the sound pressure squared at a distance r (at least several source diameters) on a selected axis of the sound source to the sound pressure squared that would be produced at the same point if the source were non-directional, but radiating the same total acoustic power. Thus, before Q(f) can be evaluated, it is necessary to determine the power radiated by the source at the given frequency.

For a directional source, the total radiated power can be obtained by measuring or estimating the sound pressure levels at all points on a sphere surrounding the source. The radius of the sphere should be several source diameters. Usually it is only possible to make measurements at selected points. The sound pressure levels at other points must then be estimated by interpolation between measured levels or by making use of geometrical symmetry in the source. To insure that the error in the measurements is very small, a sufficient number of readings should be taken so that the maximum variation between adjacent readings is no greater than about 10 decibels. The sphere (or hemisphere) over which measurements are taken should then be divided into a number of areas, one area corresponding to each measurement position used. The power passing through each area, S in sq ft, at the frequency or in the frequency band for which the sound pressure level has been determined, is

$$W_s(f) = \frac{372}{\rho c} \times 10^{-14} \, S \, \text{antilog}_{10} \left[\frac{SPL}{10}\right] \text{ watts} \qquad (3.6)$$

The sum of all the powers so determined represents the total acoustic power, in watts, radiated by the source at the given frequency or in the given frequency band for which the measurements were made. By means of Eqs. (3.2) or (3.4) this power can be converted to a power level. If measurements are made for various frequencies or frequency bands, the power or power level can be plotted as a function of frequency. Finally the sum of these individual powers yields a figure for the total power radiated by the source for all frequencies or frequency bands. This total power can similarly be converted by means of Eqs. (3.2) or (3.4) to a total power level. The work can be done more conveniently when all the measurement areas are equal.

The calculation for power can be greatly simplified by computing an average sound pressure level in decibels as the average of the SPL's when equal-area measuring sections are used. Unless all of the sound pressure levels are equal some error will be present. The value computed in this fashion will always be lower than the true average of the sound pressure levels (see Chapter 2). For example, if there is a variation as great as 10 db among four to eight measurements, the maximum error will be -2.6 db. Thus the following rule should be adopted. If the spread of the measured values of the sound pressure level is approximately 10 db, and if the decibel readings are averaged, add one decibel to the average. The resulting average will probably not be in error more than one db. If the spread of measured values is 5 db or less, and decibel readings are averaged, no correction will be necessary since the error will always be less than one db.

Quite often it will not be possible to obtain readings over a surface completely surrounding the source, and at a distance of several source diameters. However, the radiation pattern produced by the sound source will frequently exhibit some form of symmetry which permits the problem to be simplified.

An example of an approximately correct simplification, which might be used in practice, is shown in Fig. 3.2. The source of sound is the open end of an engine test cell. The opening, the center of which is at point C, is in the vertical plane, so that the most intense sound is radiated in a horizontal direction. The plane xy of the diagram represents the surface of the earth. If the opening were circular, the

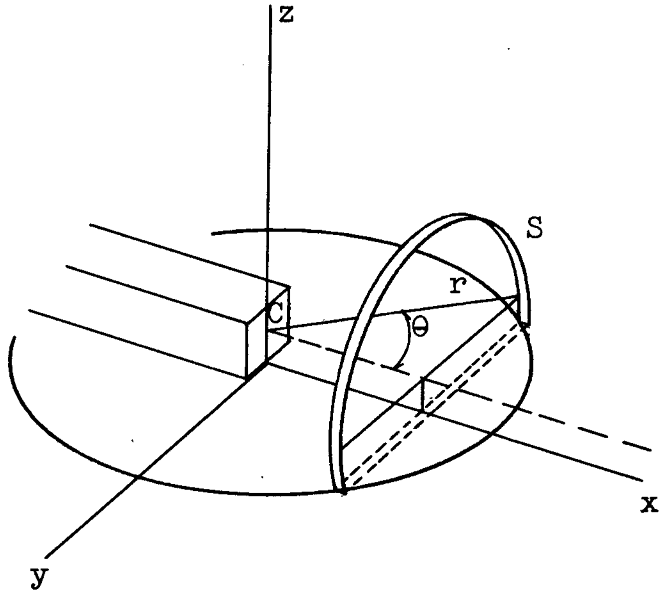


Figure 3.2

Sketch showing possible arrangement of survey-area geometry when the radiation is approximately symmetrical about an axis parallel to the ground.

radiation would be closely symmetrical about the perpendicular to the opening, which is indicated by the broken line. This symmetry will be approximated even with the actual opening, which is square. Therefore, it may be assumed that the SPL is nearly constant for all points on an arc defined by constant values of r and of the angle Θ as shown in the figure. An element of area S for the survey may be taken as the area, on the surface of a sphere of radius r, which is bounded by two arcs corresponding to different constant values of the angle Θ and by the xy plane. Since the SPL is nearly constant over each such element of area, provided that the width of the element in terms of the angle Θ is sufficiently small, one measurement of SPL will suffice for each element. This measurement may be made at a convenient position several feet above the surface of the earth. Consequently, the actual survey points are located a few feet

above an arc on the ground which defines a constant distance <u>r</u> from the center of the opening.

An approximation which is somewhat more accurate for the situation of Fig. 3.2 is obtained by subdividing each element of area as defined above into a number of segments which become new, smaller elements of area. The survey positions are then much greater in number, because the microphone must be several positions along the arc. This is difficult in practice, because a number of the survey positions are at considerable elevations above the ground. It is necessary to survey only half of the arc, however, for it may be assumed that the distribution of SPL is symmetrical about the xz plane.

Once the power level of the source has been determined as a function of frequency, the directivity factor can be evaluated. For a directional source, Eq. (3.3) must be modified to read

$$\mathrm{SPL} = \mathrm{PWL} - 10 \log S_s + 10 \log Q(f) + [10 \log \rho c - 15.7] \tag{3.7}$$

where S_s is the area in sq ft of the spherical surface that encloses the source. If the source is on a ground plane, S_s is the area of a hemispherical surface.

The directivity index is defined as

$$\mathrm{DI}(f) = 10 \log Q(f) \tag{3.8}$$

Therefore,

$$\mathrm{DI}(f) = \mathrm{SPL} - \mathrm{PWL} + 10 \log S_s - [10 \log \rho c - 15.7] \tag{3.9}$$

Obviously the value of Q(f) or DI(f) will depend upon the direction of the selected axis along which the sound pressure level is measured relative to a chosen reference axis with the source as origin. Thus Q(f) or DI(f) is usually identified with the two angles of the spherical coordinate system, θ and ϕ, taken with respect to a principal or selected reference axis.

3.4 Reflecting Surfaces

In the previous sections it has been assumed that the source is in a free field region or, in special cases, on

a large plane surface. Quite often, however, data concerning a source are obtained in the presence of one or more reflecting surfaces. In order to evaluate the characteristics of the source, corrections must be made for these surfaces. It will be assumed that the pressure of reflecting surfaces does not alter the properties of the source. This assumption will be satisfied in practically all cases of interest.

If a sound wave is incident upon a rigid surface at any angle, the pressure at the surface will be doubled, that is, the SPL will be 6 decibels greater. Interference between the incident and the reflected sound wave will result in a standing-wave pattern. The standing-wave pattern consists of periodic variation of sound pressure in space from zero to twice the pressure in the incident sound wave. If the surface upon which the sound is incident is not rigid, the problem of reflection becomes very complicated and is a function of the acoustical impedance of the surface as well as the angle of incidence of the sound. In the practical specification of noise sources, it is often permissible to assume that the principal reflecting surfaces are rigid, and a correction of -6 db is applied to measurements made in the vicinity of a surface. In case the area of the surface does not subtend many wavelengths of sound, diffraction problems will arise which can only be solved by a qualified acoustical engineer. When enough surfaces are present to constitute an enclosure, the problem takes on the form discussed in the next section.

3.5 Diffuse Fields

If the noise source is within an enclosure, its sound field will be greatly affected by the enclosure walls, and it is necessary to reconsider the measurement techniques. An enclosure can be looked upon as a compound system of reflecting surfaces. The walls of an enclosure will reflect a portion of the sound that strikes them, and if the enclosure is large, reverberant, and irregular, a large number of reflections will occur. Essentially a very large number of standing-wave patterns will be established such that, in the steady state, a relatively uniform sound level will be created throughout the room except in the vicinity of the sound source. Near the sound source the 6 db decrease of SPL for each doubling of distance will still be observed. The sound field established away from the source is called a diffuse field.

Assume that the sound source is located at a sufficient distance from all room surfaces that the sound pressure level in the vicinity of a surface is due primarily to the reverberant sound. Then the following equation gives the average SPL at a distance r from the source: 1/

$$SPL = PWL + 10 \log \left[\frac{Q(f)}{4\pi r^2} + \frac{4}{R}\right] + [10 \log \rho c - 15.7] \quad (3.10)$$

where

- r = distance in ft from sound source
- Q = directivity factor
- $R = \frac{\alpha S}{1 - \alpha}$ (at a given frequency)
- S = the total area of the bounding surfaces of the room in sq ft
- α = average absorption coefficient of the surfaces of the room.

In detail

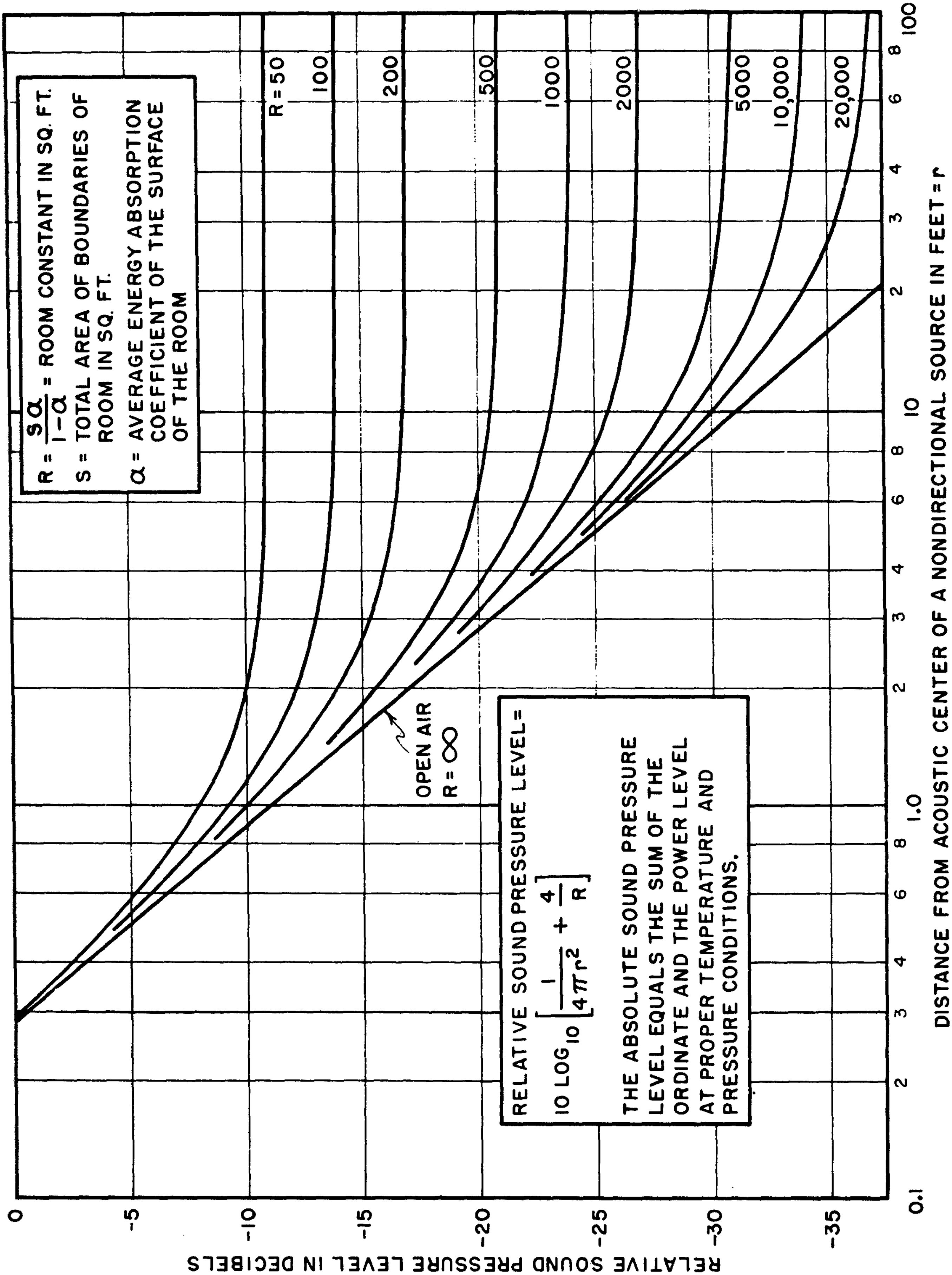
$$\alpha = \frac{\alpha_1 S_1 + \alpha_2 S_2 + \alpha_3 S_3 + \ldots\ldots\ldots\ldots\ldots + A_s + A_p}{S}$$

where

- α_1, α_2.......= absorption coefficients of individual surfaces
- S_1, S_2.......= respective area of those individual surfaces
- A_s = absorption of all seats and furnishings in the room, not occupied by people
- A_p = absorption of people in the room.

In the case of a partial enclosure, α is considered to be unity for the open areas.

For a uniform spherical source the directivity factor, Q, is unity. For such a source a plot of the variation of sound pressure level as a function of distance in feet from the center of the point surface is given in Fig. 3.3. For perfectly absorbing walls ($\alpha = 1$, and R infinite), the sound pressure level decreases 6 decibels for each doubling of distance. The less absorption there is at the surfaces of the enclosure, the smaller the distance one must go from the source to measure a sound pressure level that is independent of distance. For example, if R equals 100, a diffuse and uniform sound field will exist within 10 ft of the source. For that part of the curve where the sound pressure is constant,



R = $\frac{S\alpha}{1-\alpha}$ = ROOM CONSTANT IN SQ. FT.
S = TOTAL AREA OF BOUNDARIES OF ROOM IN SQ. FT.
α = AVERAGE ENERGY ABSORPTION COEFFICIENT OF THE SURFACE OF THE ROOM
R=50
100
200
500
1000
2000
5000
10,000
20,000
OPEN AIR
R = ∞
RELATIVE SOUND PRESSURE LEVEL=
$10 \log_{10} \left[\frac{1}{4\pi r^2} + \frac{4}{R}\right]$
THE ABSOLUTE SOUND PRESSURE LEVEL EQUALS THE SUM OF THE ORDINATE AND THE POWER LEVEL AT PROPER TEMPERATURE AND PRESSURE CONDITIONS.
0
-5
-10
-15
-20
-25
-30
-35
RELATIVE SOUND PRESSURE LEVEL IN DECIBELS
0.1
2
3
4
6
8
1.0
2
3
4
6
8
10
2
3
4
6
8
100
DISTANCE FROM ACOUSTIC CENTER OF A NONDIRECTIONAL SOURCE IN FEET = r

the following simplified relation holds.

$$SPL = PWL + 6.0 - 10 \log R + [10 \log \rho c - 15.7] \qquad (3.11)$$

As in the previous equations, the quantity in brackets is a fraction of a decibel at usual room temperature and pressure. Equation (3.11) applies when r^2 is large compared to $R/16\pi$. If r^2 is small compared to $R/16\pi$, the inverse distance relation given by Eq. (3.5) can be used.

Quite often the reverberation time of the enclosure will be known instead of the room constant, R. We can then substitute for R the expression

$$R = \frac{0.05\ V}{T}\left[1 + \frac{0.025\ V}{TS}\right] \qquad (3.12)$$

where

V = volume of room in cu ft
S = total surface of room in sq ft
T = reverberation time in seconds.

For a directional source, Q(f) will not be equal to one, and the variation of sound pressure level as a function of distance from the center of the source will depend upon the axis of the sound source along which distance is measured. Figures 3.4 and 3.5 illustrate the situation for a directional source with various values of Q, the directivity factor. Again, there is a distance from the source beyond which the sound pressure level will be constant.

Measurements on a sound source located within an enclosure require considerable care. Measurements should be made at a distance of several source diameters from the source. The relationship between r^2 and $R/16\pi$ will determine which of Eqs. (3.5), (3.10), (3.11) should be used. However, if the value of R is not known accurately, r^2 should be taken small compared to the approximate value of $R/16\pi$. The directivity factor of a sound source is best determined by outdoor measurements where there is no contribution from reflected

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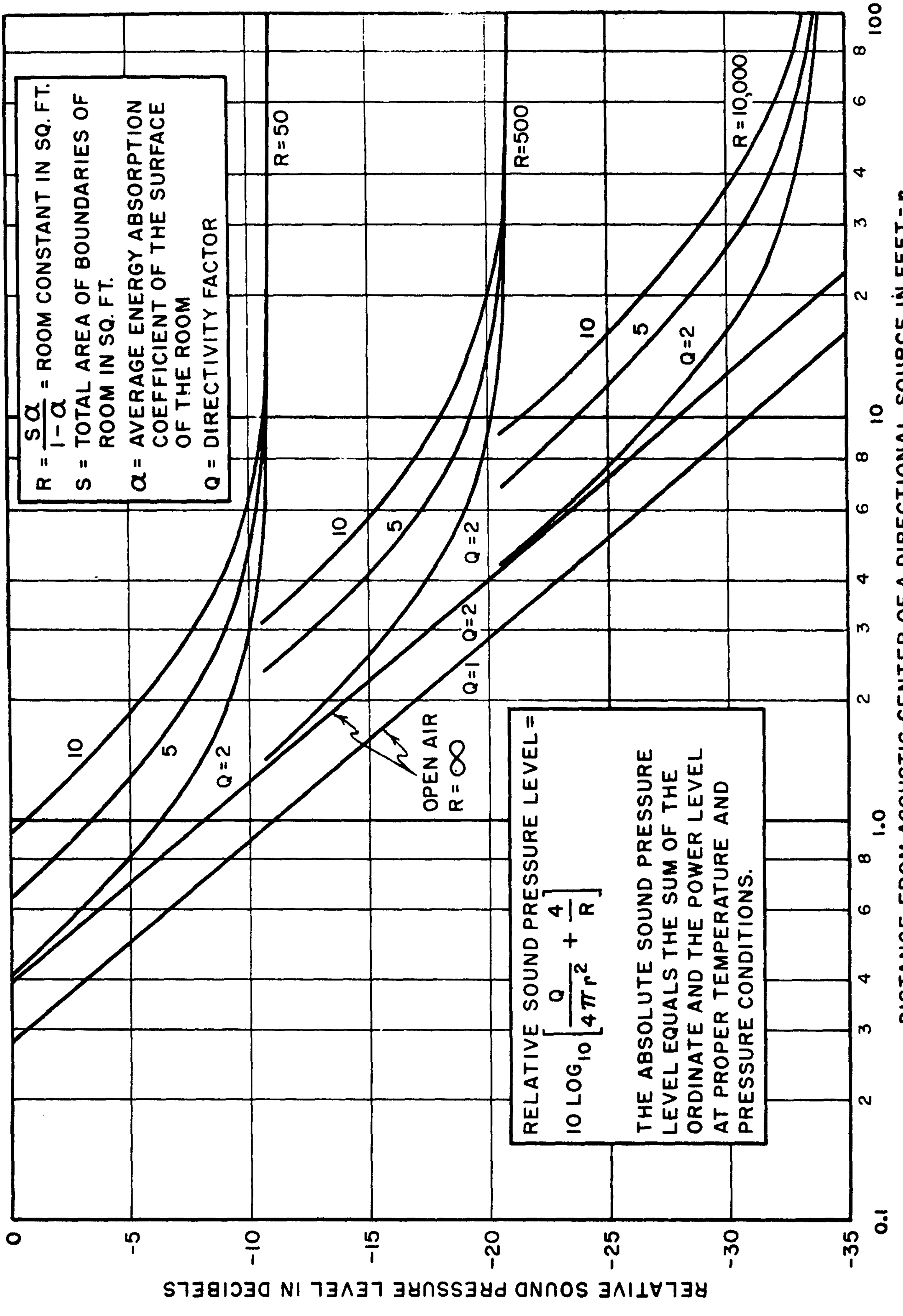
Figure 3.3

Relative SPL as a function of distance from a non-directional source in an enclosure. See Eq. (3.10).



R = $\frac{S\alpha}{1-\alpha}$ = ROOM CONSTANT IN SQ. FT.
S = TOTAL AREA OF BOUNDARIES OF ROOM IN SQ. FT.
α = AVERAGE ENERGY ABSORPTION COEFFICIENT OF THE SURFACE OF THE ROOM
Q = DIRECTIVITY FACTOR
R=50
R=500
R=10,000
10
5
Q=2
Q=1
OPEN AIR
R = ∞
RELATIVE SOUND PRESSURE LEVEL =
$10 \log_{10}\left[\frac{Q}{4\pi r^2} + \frac{4}{R}\right]$
THE ABSOLUTE SOUND PRESSURE LEVEL EQUALS THE SUM OF THE ORDINATE AND THE POWER LEVEL AT PROPER TEMPERATURE AND PRESSURE CONDITIONS.
RELATIVE SOUND PRESSURE LEVEL IN DECIBELS
0
-5
-10
-15
-20
-25
-30
-35
0.1
2
3
4
6
8
1.0
2
3
4
6
8
10
2
3
4
6
8
100
DISTANCE FROM ACOUSTIC CENTER OF A DIRECTIONAL SOURCE IN FEET = r

sound. If measurements must be made indoors, the sound pressure level at the selected measuring points should be at least 8 decibels greater than the average sound pressure level measured at more distant points where reflected sound predominates. This will insure that the reflected sound does not appreciably contribute to the readings. However, the measuring point should still be at a distance of several source diameters from the sound source.

In a highly reverberant room it may be impossible to satisfy the above conditions unless (1) a sufficient amount of sound-absorbing material is introduced to decrease the reverberation time, T, and increase the room constant, R, or (2) a large enclosure lined with acoustical absorbing material on non-parallel surfaces is built around the source. The material should have a thickness approximately 0.08 times the longest wavelength at which measurements are to be made. The determination of directivity indices and power levels as a function of frequency is then the same as described in Sec. 3.3

3.6 Noise Source in a Duct

When the total power output of a sound source is distributed nearly uniformly over an area of S sq ft, and when sound energy can flow through this area without reflection, a simple relationship exists between the area, the PWL, and SPL, as indicated by Eq. (3.3). This equation is repeated for convenience.

$$\text{SPL} = \text{PWL} - 10 \log S + [10 \log \rho c - 15.7] \qquad (3.3)$$

This relationship affords a convenient basis for finding the power level of a source when it is known that substantially all the power must pass through a duct, as illustrated in Fig. 3.6. Here the source is confined to an enclosure which

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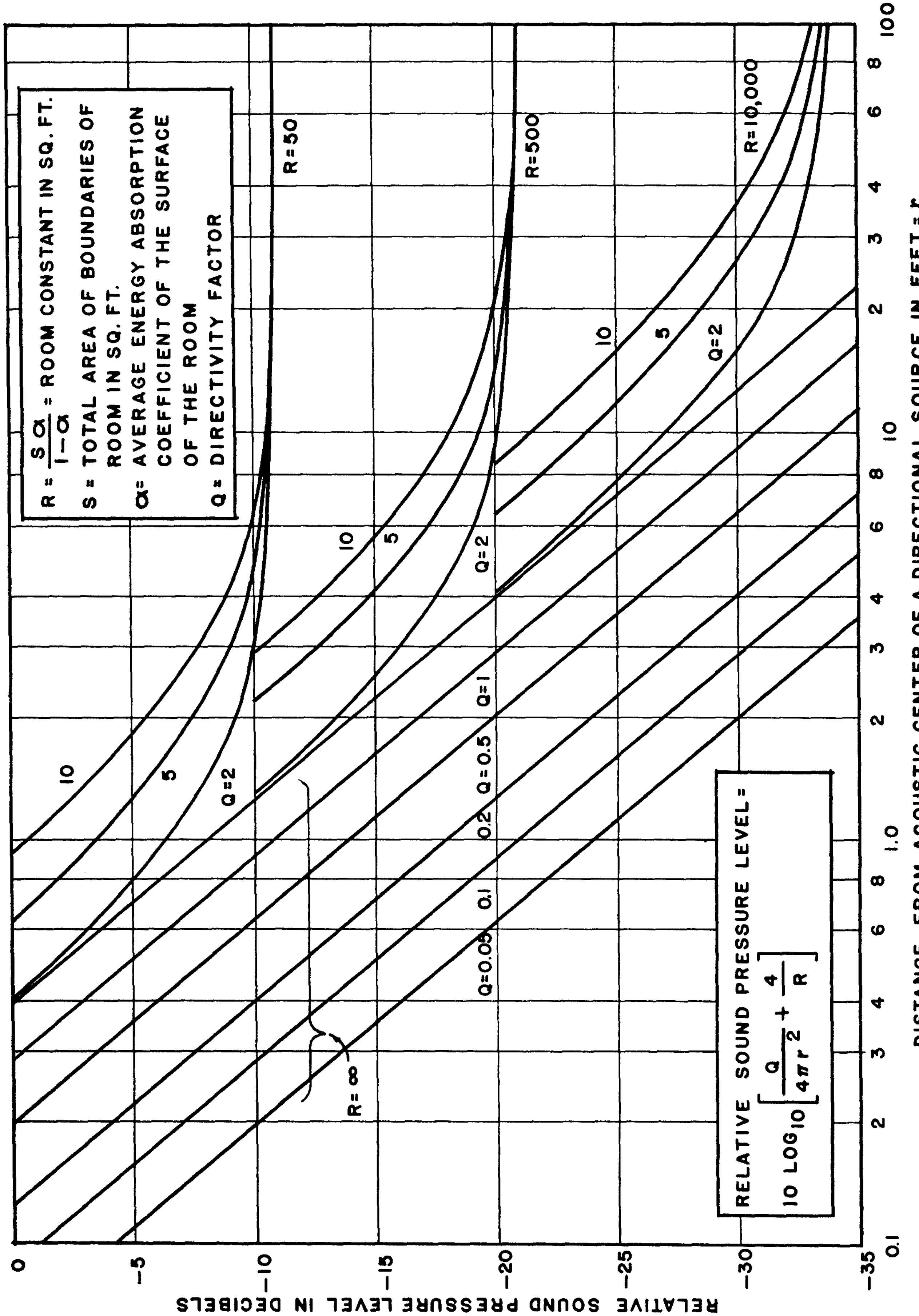
Figure 3.4

Relative SPL as a function of distance from a source in an enclosure, for various values of the directivity factor Q greater than unity. See Eq. (3.10).



$R = \frac{S\alpha}{1-\alpha}$ = ROOM CONSTANT IN SQ. FT.
S = TOTAL AREA OF BOUNDARIES OF ROOM IN SQ. FT.
α = AVERAGE ENERGY ABSORPTION COEFFICIENT OF THE SURFACE OF THE ROOM
Q = DIRECTIVITY FACTOR
RELATIVE SOUND PRESSURE LEVEL =
$10 \log_{10}\left[\frac{Q}{4\pi r^2} + \frac{4}{R}\right]$
R=50
R=500
R=10,000
R=∞
Q=2
5
10
Q=0.05
0.1
0.2
Q=0.5
Q=1
RELATIVE SOUND PRESSURE LEVEL IN DECIBELS
0
−5
−10
−15
−20
−25
−30
−35
DISTANCE FROM ACOUSTIC CENTER OF A DIRECTIONAL SOURCE IN FEET = r
0.1
2
3
4
6
8
1.0
10
100

has no opening other than the duct. If the sound energy passes down the duct without reflection, the power level of the source may be found from Eq. (3.3), when the value of SPL is that measured in the duct at a location where the area is S. This assumes that there is no reduction in sound energy output from the source as a consequence of being enclosed in the duct.

The sound energy will pass down the duct without significant reflection if there are no abrupt changes in cross-sectional area, and if one of the following conditions is realized: (1) the duct is terminated in the open air, with the exit perimeter greater than the wavelength of the sound or, (2) the duct is terminated in a structure which is a nearly perfect sound absorber.

When there is an abrupt change in duct area, it is advisable to make measurements on the side of the discontinuity away from the source. Measurements should be made several duct diameters away from the discontinuity. If there is no appreciable reflection, the values SPL_1 and SPL_2, measured on opposite sides of the area change where the areas are respectively S_1 and S_2, will in the case of low flow velocities through the duct be related by Eq. (3.13):

$$10 \log (S_1/S_2) = SPL_2 - SPL_1 + 10 \log \frac{(\rho c)_1}{(\rho c)_2} \qquad (3.13)$$

In Eq. (3.13), $(\rho c)_1$ and $(\rho c)_2$ are the values of the specific acoustic resistance of air in the two positions.

Some criterion is required for estimating whether it is true that substantially all of the radiated power passes into the duct. The following rule is adequate for practical purposes, when the enclosure walls have very small transmission. If the total absorption of the enclosure in sabins (not counting the duct entrance) is less than 5 percent of the entrance area of the duct, in sq ft, it may be assumed that all sound power enters the duct. For an explanation of how to calculate the power entering the duct when the walls transmit appreciable sound power, see Chapter 10.

Figure 3.5

Relative SPL as a function of distance from a source in an enclosure, for various values of the directivity factor Q less than unity.

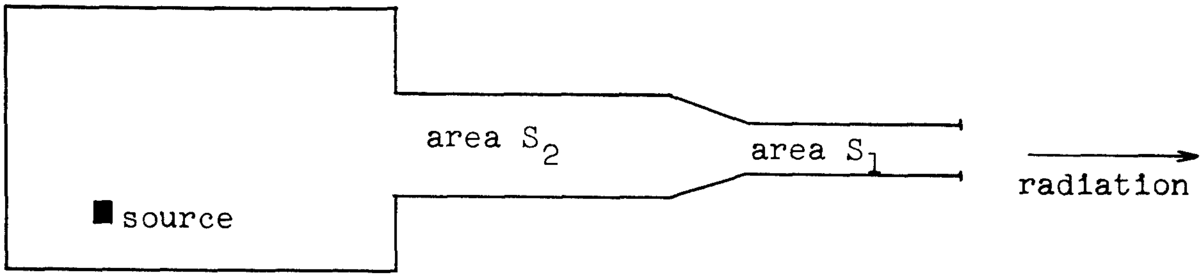


Figure 3.6

Sound source in an enclosure connected to a single duct. If the sound absorption of the enclosure walls is very small, all the sound power must pass through the duct.

The principles stated above are employed in the investigation of sources of noise within pipes. When the noise source can radiate in two directions in a long pipe, or when two equal ducts open from an enclosure, 3 db must be added to the value of PWL obtained from Eq. (3.13).

3.7 Additional Considerations

The material in the preceding sections applies when the sound source is very small, or when the observed values of SPL are obtained at points outside the region of the near field of the source. Near-field effects are avoided when the distance from the source to the point of measurement is several times the largest dimension of the source.

Practical sound sources are frequently distributed over a region of significant size. An example is a turbo jet engine, which delivers radiation from front and rear openings. The behavior of the SPL as a function of distance, when the source is distributed, has the general features shown in Fig. 3.7.

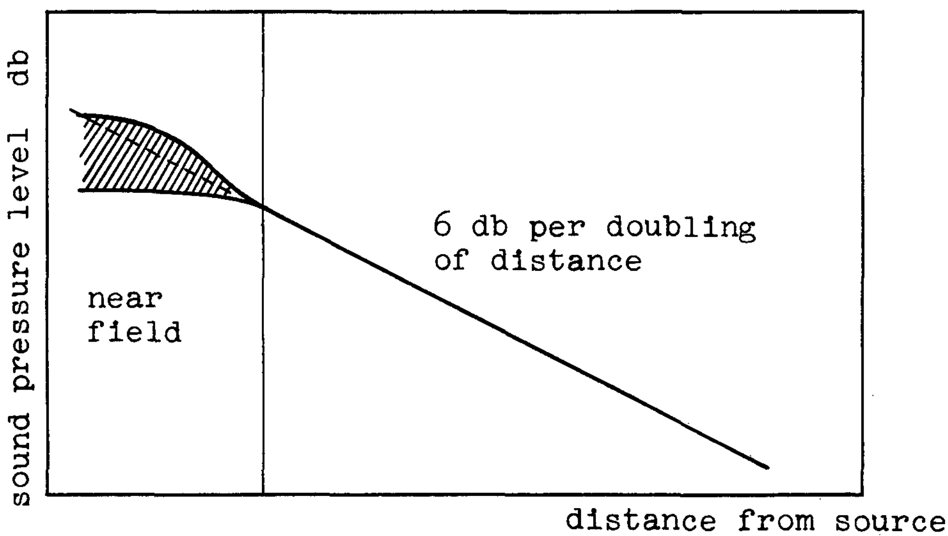


Figure 3.7

Variation of SPL as a function of distance from the center of a distributed sound source in the open air.

When a distributed source is located within an enclosure, the variation of SPL with distance may be of the form shown in Fig. 3.8. Here it is possible to distinguish the near-field region, a region in which the inverse-square relation holds approximately, and an outermost region in which the SPL is that of the diffuse field in the room and is nearly independent of distance. The diffuse region is found where r^2 is much greater than $R/16\pi$.

Whenever the characteristics of a noise source are to be measured, the variation of sound pressure level with distance should be studied by means of several measurements. Thus one can determine the region where the inverse distance relation holds, and the region of uniformity of the pressure field (no prominent standing-waves) at a distance from a source situated in an enclosure.

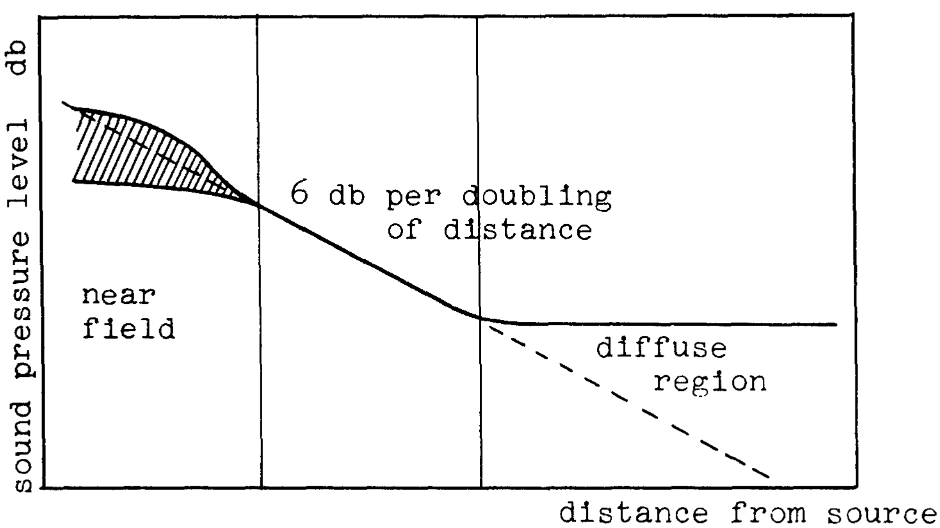
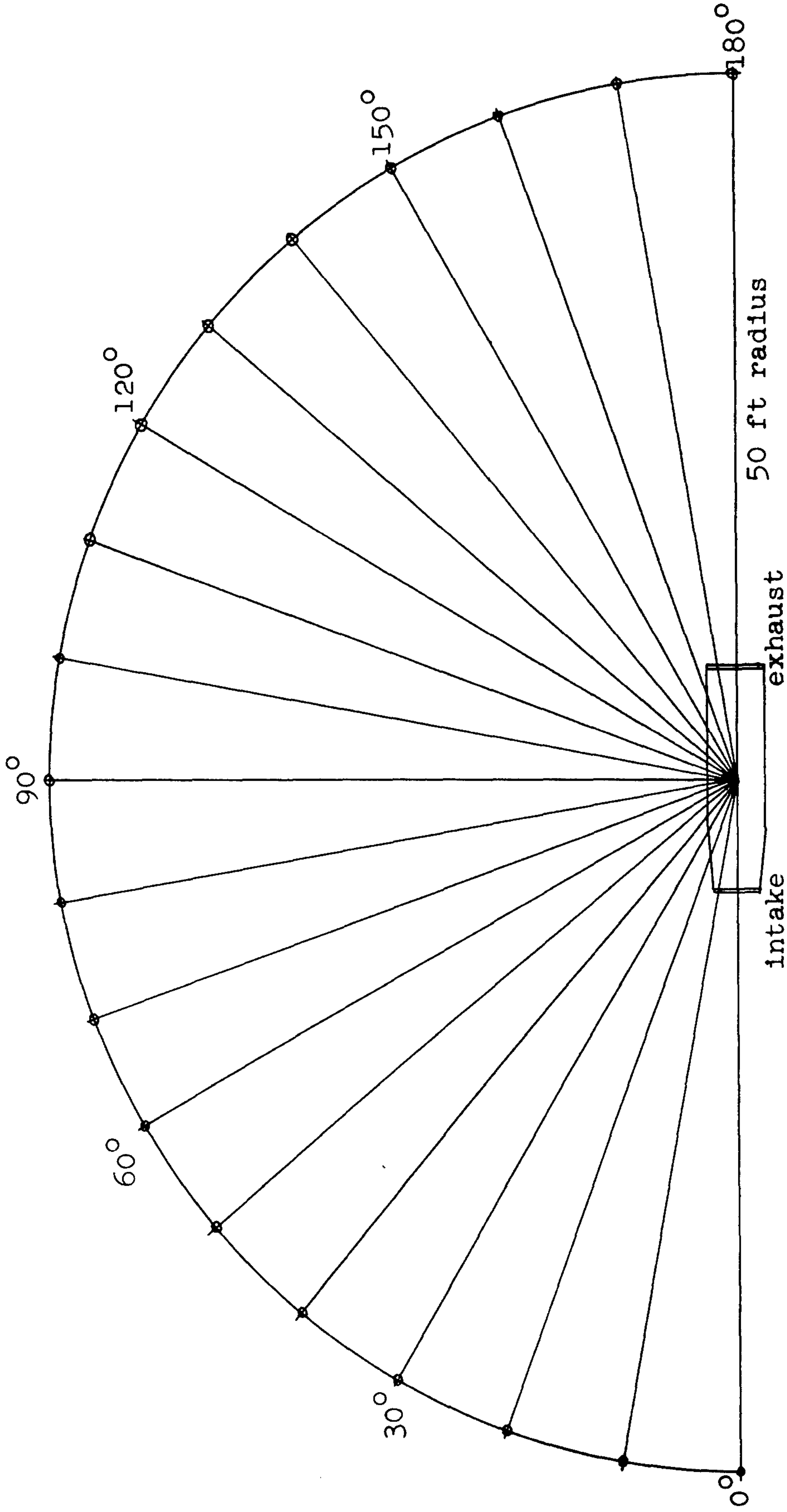


Figure 3.8

Variation of SPL as a function of distance from the center of a distributed sound source in an enclosure with partially reflecting walls.



PLAN VIEW

Related to this variation of sound pressure level with distance from the source, is the manner in which the directivity of the source may vary with distance. This effect, which is discussed in Chapter 12, may be due to reflections from surfaces, meteorological conditions, etc. Thus the directivity of the source should be similarly studied to determine the distance at which it approaches an invariant condition.

Finally, there may be problems which do not warrant a full analysis of source properties. For example, in studying the noise at large distances from an engine in a closed test cell, knowledge of the total power output of the source without regard to directivity is at times all that is desired. Or, if one wishes to measure at a residence the disturbance that results from passing highway vehicles, measurements at a fixed roadside point, with no attempt to arrive at the power output and directivity of the sources, will usually suffice.

3.8 Numerical Example: Survey of a Sound Source

The numerical example corresponds to the survey plan shown in Fig. 3.9. The source is a jet engine located outdoors on level ground. The survey points are located on a circle of 50 ft radius as indicated in the figure, and on the same level as the center of the engine. The following assumptions are made:

1. Surveys made at greater distance from the source have indicated that the 50 ft circle lies outside the near-field region of the source.

2. The source has symmetry of the kind discussed in Sec. 3.3 so that survey areas as shown in Fig. 3.2 may be used.

3. The survey areas may be approximated as bands about a hemisphere. Since the center of the source is not exactly at ground level the areas extend slightly more than halfway around a sphere.

Figure 3.9

Arrangement of survey points for the numerical example of Sec. 3.8.

Each survey point may be regarded as the angular center of a survey zone. Thus the survey point at 0° is at the center of a zone which extends from 355° to 5°; the survey point at 10° is at the center of a zone which extends from 5° to 15°, etc. If θ_1, θ_2 are the angular limits of a particular zone, the area of this zone is

$$S = \pi r^2 (\cos \theta_1 - \cos \theta_2) \qquad (3.14)$$

A convenient approach to the problem of computing both the source power and the directivity is to find the SPL corresponding to the average of the squared pressure. This quantity, which will be designated as SPL_{av}, is not in general equal to the average of the decibel SPL values of the survey points although the average of the decibel values may be used when all survey areas are of equal size, subject to the restrictions stated in Sec. 3.3. The value of SPL_{av} is given by the relation

$$(\text{Total area}) \times \text{antilog} \left(\frac{SPL_{av}}{10}\right) = \sum_j (S_j) \text{ antilog } \frac{(SPL)_j}{10} \qquad (3.15)$$

where (total area) refers to the entire area through which sound radiation passes, and the summation refers to all survey areas. The sum of the survey areas S_1, S_2, etc. must equal the total area.

The total power radiated is found by using the value of SPL_{av} in Eq. (3.6).

The data for the numerical example are shown in Table 3.1. The values do not correspond to any actual measurements. The table shows the values of SPL observed at the various survey points, the areas of the corresponding zones, and the quantity S antilog (SPL/10) for each zone. The sum of the latter quantity is 1.38×10^{-13}. By Eq. (3.15), the value of SPL_{av}, which is the SPL that would be produced by uniform hemispherical radiation of the observed power, is

$$SPL_{av} = 89.5 \quad \text{db}.$$

By Eq. (3.6), the total power in watts is

$$W = \left(\frac{372 \times 10^{-14}}{c}\right) \times 1.38 \times 10^{13} .$$

TABLE 3.1

DATA FOR NUMERICAL EXAMPLE

Angular location of measurement station, degrees	Zone area, sq ft S	SPL db	S antilog (SPL/10) Units 10^{11} sq ft	DI, db
0	30	94	0.8	4.5
10	240	93	4.8	3.5
20	465	94	11.6	4.5
30	685	92	11.0	2.5
40	890	90	8.9	0.5
50	1040	87	5.2	-2.5
60	1090	86	4.4	-3.5
70	1290	86	5.1	-3.5
80	1350	86	5.4	-3.5
90	1370	86	5.5	-3.5
100	1350	86	5.4	-3.5
110	1290	86	5.1	-3.5
120	1090	87	5.5	-2.5
130	1040	87	5.2	-2.5
140	890	89	7.1	-0.5
150	685	93	13.7	3.5
160	465	94	11.6	4.5
170	240	96	9.6	6.5
180	30	98	1.9	8.5

If the value of ρc is 41 rayls, the power radiated is

$$W = 1.25 \text{ watts.}$$

The power level is

$$PWL = 10 \log W + 130.0 = 131 \text{ db .}$$

The directivity index for each measurement position is simply the difference between the observed SPL for that position and the value SPL_{av}. The DI values are included in Table 3.1.

The data given are assumed to apply to the 1200-2400 cps frequency band. A similar set of data must be obtained and processed for each frequency band in order to give a complete description of the source.

Attempts should be made to correlate the important properties of the source such as power spectrum, total power, and the distribution as functions of the characteristic physical parameters of operation of the source. These parameters may be numerous; for example, in the operation of a propeller one must consider horsepower, tip speed, number and shape of blades, etc. If possible, these parameters should be reduced to a few important parameters for optimum use in engineering design charts. As an example, over a wide range of operating conditions, the total acoustic power from a ram jet engine can be expressed approximately as a function of the fuel consumption rate. Additional plots could include octave-band spectra of the total power and directivity patterns in separate frequency bands. The final analysis, interpretation, and use of the data are limited only by the ingenuity and imagination of the acoustical engineer.

References

(1) Hopkins, H. F. and Stryker, N. R., "A Proposed Loudness-Efficiency Rating for Loudspeakers and the Determination of System Power Requirements for Enclosures", Proc. Inst. Radio Eng. 36 315-335 (1948).

CHAPTER 4

AIRCRAFT PROPELLERS AND RECIPROCATING ENGINES

4.1 Propeller Noise

Some of the operating parameters which may reasonably be considered in the description of noise from propellers are the following:

Input horsepower
Thrust
Blade tip speed
Angle of attack
Propeller diameter
Number of blades
Blade shape (especially tip shape).

Fortunately it has been found that an engineering description of propeller noise, which is adequate to cover a wide range of normal operating conditions, can be given in terms of (1) blade tip speed; (2) input horsepower per blade; (3) number of blades.

The published results1/ of an extensive series of noise measurements in the cabins of airplanes in flight suggest the following useful generalizations concerning the overall noise level generated by a propeller of given type:

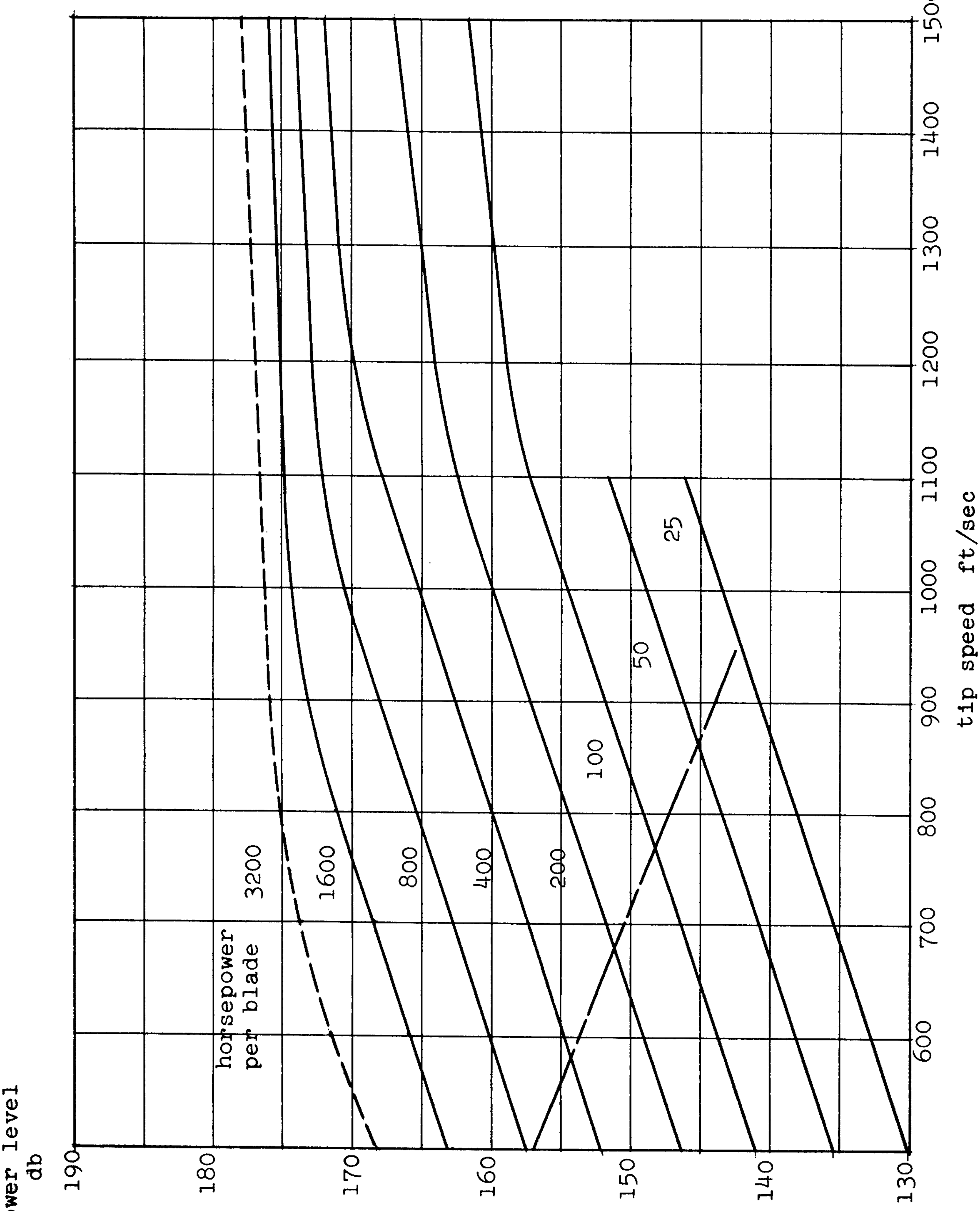
1. The overall SPL due to the propeller increases by approximately 5.5 db for each doubling of the input horsepower per blade.

2. The overall SPL due to the propeller increases by approximately 2.7 db for each increase of 100 ft/sec in tip speed.

3. The overall noise energy is proportional to the number of blades, when horsepower per blade and tip speed are fixed.

4. The overall propeller SPL is greater by about 3 db for a blunt-tipped propeller than for a propeller with fine pointed blade tips.

Because of the complications introduced by proximity to the source and by the presence of cabin walls, these results for


power level
db
190
180
170
160
150
140
130
600
700
800
900
1000
1100
1200
1300
1400
1500
tip speed ft/sec
horsepower
per blade
3200
1600
800
400
200
100
50
25

aircraft in flight do not lead to values of the power level of the propellers as a noise source. Moreover, these results were derived entirely from data for blade tip speeds less than Mach 1.0.

Overall Power Level Chart. Since the time of publication of Ref. (1), results have become available from several propeller noise tests in which the acoustical environment (partially enclosed test cell or outdoor test cell) is simple enough to allow computation of the power level of the propeller as a noise source. Some of these later results also include data for blade-tip speeds as high as Mach 1.3. Results of the later measurements have been combined with conclusions 1-4, cited above, to yield the propeller noise chart shown in Fig. 4.1. This chart, which applies directly to propellers with pointed tips, indicates the overall power level of a 3-bladed propeller as a distant noise source when values of blade-tip speed and of horsepower input per blade are supplied. The corrections for other numbers of blades are indicated on the chart. The left-hand straight-line portions of the curves in Fig. 4.1 are drawn to conform to conclusions 1 and 2 of Ref. (1). These conclusions are supported also by the more recent data. The reduced slopes which appear wherever the various curves of Fig. 4.1 extend into the right-hand portions of the chart indicate an experimental finding that the power level, at constant input horsepower, increases less rapidly with increasing tip speed when the tip speed exceeds a critical value which lies near Mach 1.0. Since this effect is more pronounced at high values of blade horsepower, it may be further stated that at large values of tip speed and blade horsepower the acoustical power level becomes less sensitive to each of these parameters.

Figure 4.1

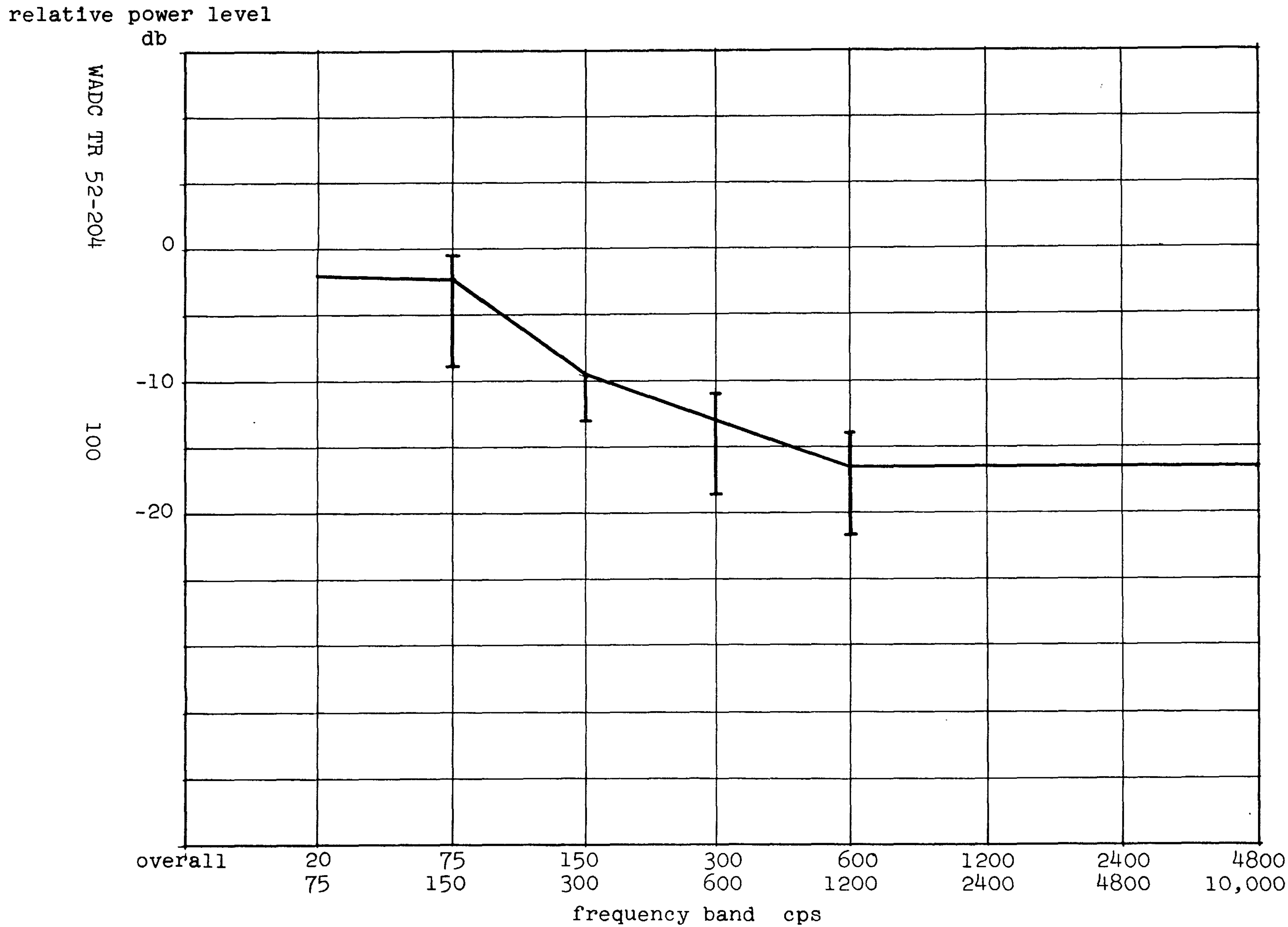
Chart for estimating power level of propeller noise. Proceed upward from the desired tip-speed value, on the horizontal scale, to the curve representing proper horsepower per blade (interpolate if necessary). Read power level directly on the vertical scale for a 3-blade propeller. Add the following corrections for other numbers of blades; 2 blades, -1.8 db; 4 blades, 1.3 db; 5 blades, 2.2 db; 6 blades, 3.0 db. Add 3 db to the result if the blade tips are blunt.

The use of the chart of Fig. 4.1 may be illustrated by an example. Suppose that it is desired to find the overall sound pressure level, on the axis of a test cell, which will exist under the following conditions:

Three-blade propeller, diameter 12 ft.
1500 horsepower, total input.
Tip speed, 1200 ft/sec.
Blunt tips.
Test cell has open ends of full area; no acoustic absorption.
Point of observation is outdoors at 600 ft distance.

Enter the chart of Fig. 4.1 with the values 500 horsepower per blade, and 1200 ft/sec. The chart indicates an overall acoustical power level (PWL) of approximately 171 db. Since the propeller has blunt tips, apply conclusion 4 of Ref. (1) and correct the result by 3 db, to obtain an overall PWL of 174 db. Under normal conditions this is approximately the value of the SPL if the available power passes uniformly through an area of one ft^2. In the present case the power spreads over a larger area which, if one assumes nearly perfect ground reflection and no directional effects in azimuth, may be taken as the area of a hemisphere of radius 600 ft. This area is $2.26 \times 10^6 ft^2$. To obtain the final values of overall SPL at the point of observation, subtract from 174 db the quantity of $10 \log_{10} (2.26 \times 10^6)$. The resulting overall SPL is 110.5 db. This approximate calculation of the outdoor sound level is made on the basis of uniform spreading of sound power over a hemisphere. The directional character of the radiation from the cell openings, discussed in Chapter 12, is neglected. The error in the value of the overall outdoor SPL is not serious, because most of the propeller noise power is radiated at low frequencies, where the directional effects are relatively small.

There are several precautions to be observed in using Fig. 4.1. First, the chart should not be used in an attempt to estimate the SPL value which will be measured by a microphone placed close to the propeller (within a distance of the order of a propeller diameter or less). Since the chart was synthesized from data obtained at a distance, the chart is inapplicable to problems in which the "near field" is important. Second, it must be emphasized that the chart gives only a measure of the total acoustic power output of the source, and does not in general indicate the SPL reading to


relative power level
db
0
-10
-20
overall
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

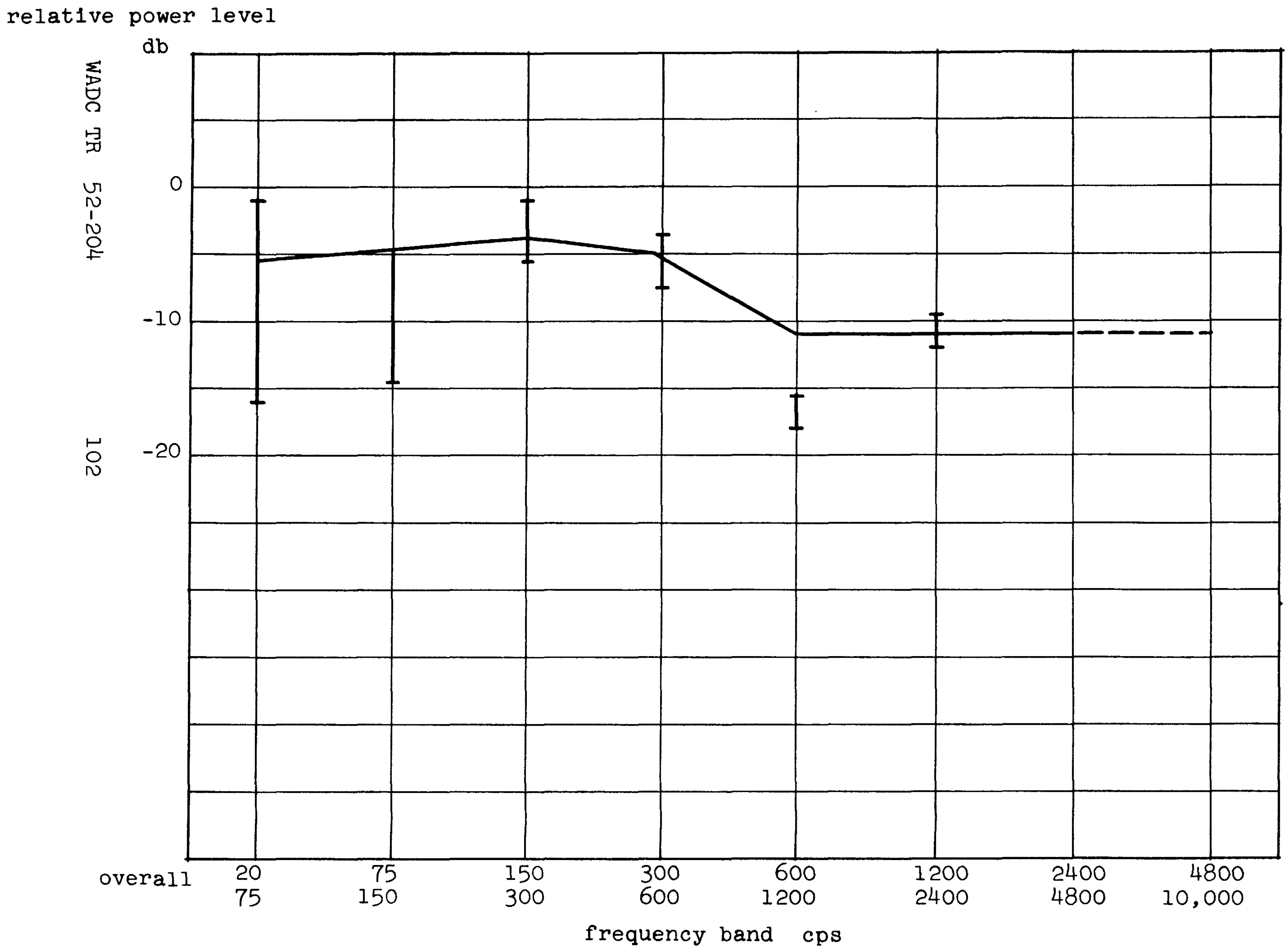
be expected at a particular point when the source is in free space. Directional effects are considered separately below. Third, the chart does not include noise generated by the engine driving the propeller.

Spectrum Charts. The spectral distribution of the acoustical power level of a propeller, as indicated by the data which were used in the derivation of the overall power level chart of Fig. 4.1, is given in Fig. 4.2 and Fig. 4.3. In each of these charts the ordinate quantity is the power level in a given octave band relative to the overall power level. The heavy vertical lines indicate the spread of the values obtained with three different propellers (diameters approximately 4 ft, 10 ft, and 17 ft). The heavy line connecting the various octave-band values is the suggested design curve, based on the available data. Figure 4.2 applies to blade-tip Mach numbers from 0.75 to 0.90, while Fig. 4.3, in which the spread of the original data is considerably greater, applies to blade-tip Mach numbers from 0.95 to 1.30. The primary difference between the design curves drawn for the two cases lies in the more rapid decline of level with increasing frequency in the case of the lower Mach range.

Most of the noise produced by a propeller occurs below 1000 cps and is propagated as a periodic wave, rich in harmonics, with the blade passage rate as the fundamental frequency. For this reason, the greatest amount of experimental information is obtained by use of a narrow band analyzer. The smaller noise energies at several thousand cps appear to consist largely of non-periodic vortex noise. If the blade passage rate is of the order of 100 per second or higher, relatively little sound pressure will be measured with ordinary filters in the 20-75 cps band; similarly if the fundamental rate is 200 per second or higher, the sound pressure in each of the two lowest bands is relatively small. The measured spectra in these cases may be approximated by sliding the design curve to the right by the required number of octaves.

Figure 4.2

Spectrum chart for propeller noise, tip Mach No. 0.75-0.90. The chart indicates the amount by which the power level for each octave band differs from the overall power level. Heavy vertical bars indicate spread of measured values.

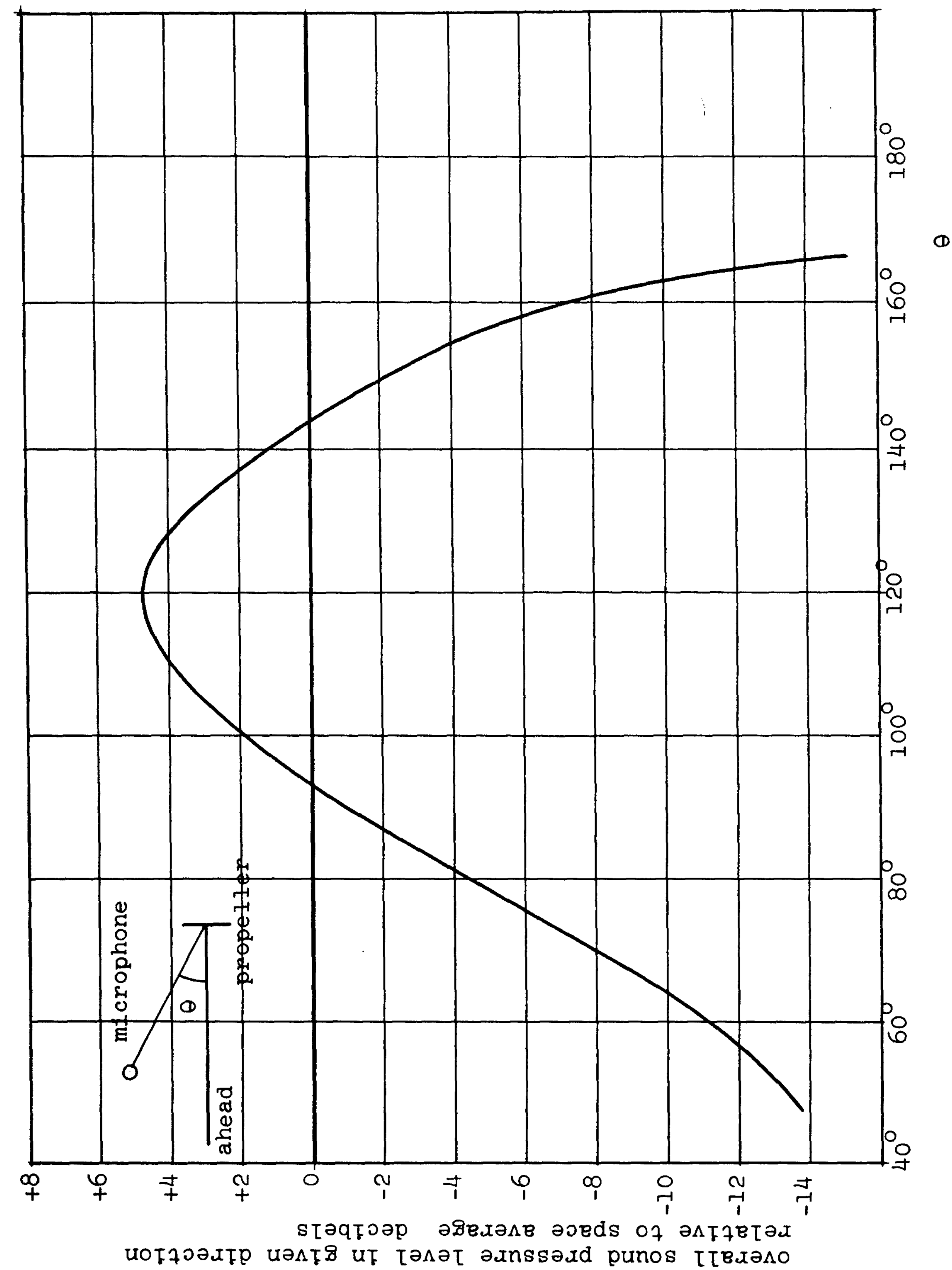

relative power level
db
0
-10
-20
overall
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

The use of the spectrum charts in Fig. 4.2 and Fig. 4.3 may be illustrated by a continuation of the example cited above. Since for the tip speed of 1200 ft/sec the tip Mach number is about 1.1, Fig. 4.3 is used here. From the given values of propeller diameter, tip speed, and number of blades, we find that the blade passage rate is 96 per second and therefore the fundamental frequency is 96 cps. The design curve of Fig. 4.3 must be moved to the right by one octave band in order to place the lowest spectrum components in the 75-150 cps band. The power levels for the various octave bands lie below the overall power level by the following amounts: 75-150 cps, 5.5 db; 150-300 cps, 4.5 db; 300-600 cps, 3.5 db; 600-1200 cps, 5.0 db; for the three highest bands, 11.0 db. The acoustical power level for each octave band may be obtained by subtracting the appropriate difference value in this group from the previously determined overall power level of 174 db. Absence of levels for the 20-75 cps band in the calculation does not mean that no acoustical treatment is required for this band; energy would be found in this band if the calculations were extended to other propellers or other operating conditions.

Directionality of Propeller Radiation. There are available a few measurements on the directionality of a propeller radiating essentially in free space. A plot for one particular case, of the directivity in decibels as a function of polar angle measured from the hub direction, is shown in Fig. 4.4. The plot refers to overall level; no breakdown of directivity by spectrum bands will be given. Information for this plot was computed from data obtained at Langley Field 2/ for a 47 in. diameter propeller operated outdoors, with the shaft horizontal. Other operating data appear in the figure. Theoretically, the directivity at large distances is a function of the polar angle only. Therefore, ground reflection should not interfere seriously with the directivity measured under these circumstances.

Figure 4.3

Spectrum chart for propeller noise, tip Mach No. 0.95-1.30. The chart indicates the amount by which the power level for each octave band differs from the overall power level. Heavy vertical bars indicate spread of measured values.


overall sound pressure level in given direction
relative to space average decibels
+8
+6
+4
+2
0
-2
-4
-6
-8
-10
-12
-14
40°
60°
80°
100°
120°
140°
160°
180°
θ
microphone
propeller
ahead
θ

While available data are insufficient to establish general design charts for the directivity of propeller noise, it is expected on the basis of the general geometrical similarity that the main features of the pattern in Fig. 4.4 will usually apply. These features are (1) the pattern is a function of polar angle only; (2) the direction of maximum intensity is 20° to 30° behind the propeller plane; (3) the maximum intensity is of the order of 5 db above the space average; (4) the axial intensity falls below the space average by as much as 15 db, with a particularly pronounced drop behind the propeller. These general conclusions are substantiated by sound surveys made on actual aircraft under ground-test conditions.3/

Basis for the Propeller Noise Charts. The propeller noise charts were derived by combining principles 1-4 of Ref. (1) with results of the following surveys:

1. Noise levels for ground-operated single-engine aircraft; microphone positions in propeller plane on either side. Propeller diameter 7.25 ft; 3 blades; tip speed 587-861 ft/sec; horsepower 245-1180.

2. Noise levels from propeller in test cell having parallel sides, open ends; microphone 200 ft distant on cell axis. Propeller diameter 16.7 ft; 4 blades; tip speed 842-1065 ft/sec; horsepower 1000-1500.

3. Noise level for 2- and 3-blade propellers of similar design, diameter 10 ft, tip speed 926-1338 ft/sec, horsepower 800-2050. Microphone locations either at 90 ft in propeller plane (for outdoor test stand) or at 200 ft on axis (for test cell with open ends).

Figure 4.4

Directivity pattern computed from overall SPL for a propeller on an outdoor test stand (Ref. 2). The directivity is the difference in db between observed SPL at a given direction and the SPL which would exist with non-directional radiation of the same total sound power.

4. Noise levels for propeller on outdoor stand 2/, various microphone positions 30 ft from hub. Diameter 47 in.; 2 blades blunt tips; tip speed 825-1430 ft/sec; horsepower 60-421.

Each of these surveys was performed with a narrow-band sound analyzer (in the case of (4), a Panoramic Frequency Analyzer). Therefore, the data consist of SPL values for a series of discrete frequencies (the blade passage frequency and its harmonics). Accordingly all engine noise components, which in general have frequencies different from those of the propeller spectrum, are easily omitted from the computations. SPL values for octave bands are derived by summing energies of the signal components of appropriate frequency. The energy of a component near a band boundary is divided by placing in the adjacent bands two energies which are 3 db lower, i.e., the sound power has been divided equally and half placed in each band.

Various corrections are applied to the observed values of SPL in order to derive source power levels. In the case of outdoor measurements, the energy is summed over all directions or, if only a single microphone position is used, a directivity correction is estimated from Fig. 4.4. Where a propeller is located within a test cell but a distant outdoor microphone is used, the assumption is made that multiple reflections within the cell cause the energy to radiate in substantially uniform spherical fashion outside the cell. (This assumption is not completely justified.) A correction is made for the attenuation of sound treatment which in one instance was placed in the test cell openings. When the microphone is placed directly on the ground, it is assumed that pressure doubling occurs and a 6 db correction is introduced. The radiation was assumed to spread over a sphere rather than a hemisphere, however. This assumption discounts the effectiveness of the ground as a reflector and leads to overall levels in the chart of Fig. 4.1 which are greater by 3 db than those which the hemispherical assumption would give, an end result which may be considered as conservative engineering practice. The overall SPL values, used in the computation of source power levels, were obtained by continuation of the process described above of adding energies in octave bands. The 3 db correction for blunt propeller tips, already mentioned, was applied to the results of survey (4).

When the power level chart as finally drawn in Fig. 4.1 is used to compute the level for each of 32 sets of experimental conditions represented in the above surveys, the chart value and the experimental value agree within 4 db in all except two cases. The chart may be described as generally correct within ± 5 db for the conditions considered. Agreement within these limits is found when the chart is compared with noise data 3/ taken in the propeller plane under take-off conditions, with a few exceptions for aircraft of 150 horsepower and less. An allowance of 5 db may be added to the chart values of overall power level for conservative engineering practice. Further uncertainty exists when octave-band power levels are predicted with the further help of Figs. 4.2 and 4.3; it is estimated that the uncertainty of octave-band levels so calculated is of the order of ± 7 db.

4.2 Noise from Aircraft Reciprocating Engines

Reciprocating engine noise has been studied less extensively than propeller noise, because the maximum noise levels produced by propeller-driven aircraft, under full-throttle conditions, are usually attributable to the propeller. The tentative generalizations given below concerning engine noise are made on the basis of a few observations (Refs. (1) and (4); also the ground airplane test cited in Sec. 4.1; also unpublished results of tests on an 800 horsepower engine in a dynamometer test cell).

1. The noise developed by a reciprocating engine is produced almost exclusively by the exhaust, with possible exceptions in cases where unusually effective mufflers are used.

2. The noise energy of the lowest-frequency exhaust component of a reciprocating engine is approximately proportional to the total power developed. Quantitatively, the power level of this exhaust component for an engine without exhaust mufflers is not less than

   Power level of lowest frequency component=

   $$122 + 10 \log_{10} \text{ (horsepower)}. \qquad (4.1)$$

   On theoretical grounds, the horsepower value used in Eq. (4.1) should include mechanical losses in the engine. However, these are usually not known. In cases where the mechanical losses are large, they must be included.

3. The lowest-frequency exhaust component of importance usually has a frequency equal to the number of exhaust discharges per second (two discharges occurring simultaneously are counted as one). This frequency is usually below 300 cps.

4. Usually the spectral distribution of noise energy is approximately as follows: The power level in the octave band containing the lowest-frequency exhaust component lies about 3 db below the overall power level. The levels in octave bands above this one decrease at about 3 db per octave of increasing frequency. No significant noise is produced in octave bands below the one containing the lowest-frequency exhaust component. These conditions may be typical of engines operated at cruising conditions, and of small engines (150 horsepower and less).

5. In the case of an engine of 800 horsepower operated at full throttle, a uniform octave-band spectrum has been observed (equal power levels in the octave band containing the lowest-frequency exhaust component and all higher octave bands). This may be typical of larger engines under full-power conditions. In this case the overall power level is about 8 db larger than that of the lowest frequency exhaust component.

6. Directional effects are much smaller for engine noise than for propeller noise. The total variation in SPL with direction is about 6 db for the lower-frequency components of engine noise. This statement probably holds for high frequencies also in the case of an isolated engine, but no detailed measurements for high frequencies are available. In the case of an engine mounted on an airplane, the high frequency directivity will be affected by shadowing produced by the airplane structure.

Simple relations for the overall power level of an engine without mufflers are obtained by combining statements 2, 4, and 5. For the case of small engines (150 horsepower or less), or engines operated under cruising conditions, the relation is

$$\text{Overall power level} = 125 + 10 \log_{10} (\text{horsepower}) . \quad (4.2)$$

For the case of a large engine operated at full load, the relation is

$$\text{Overall power level} = 130 + 10 \log_{10} (\text{horsepower}). \quad (4.3)$$

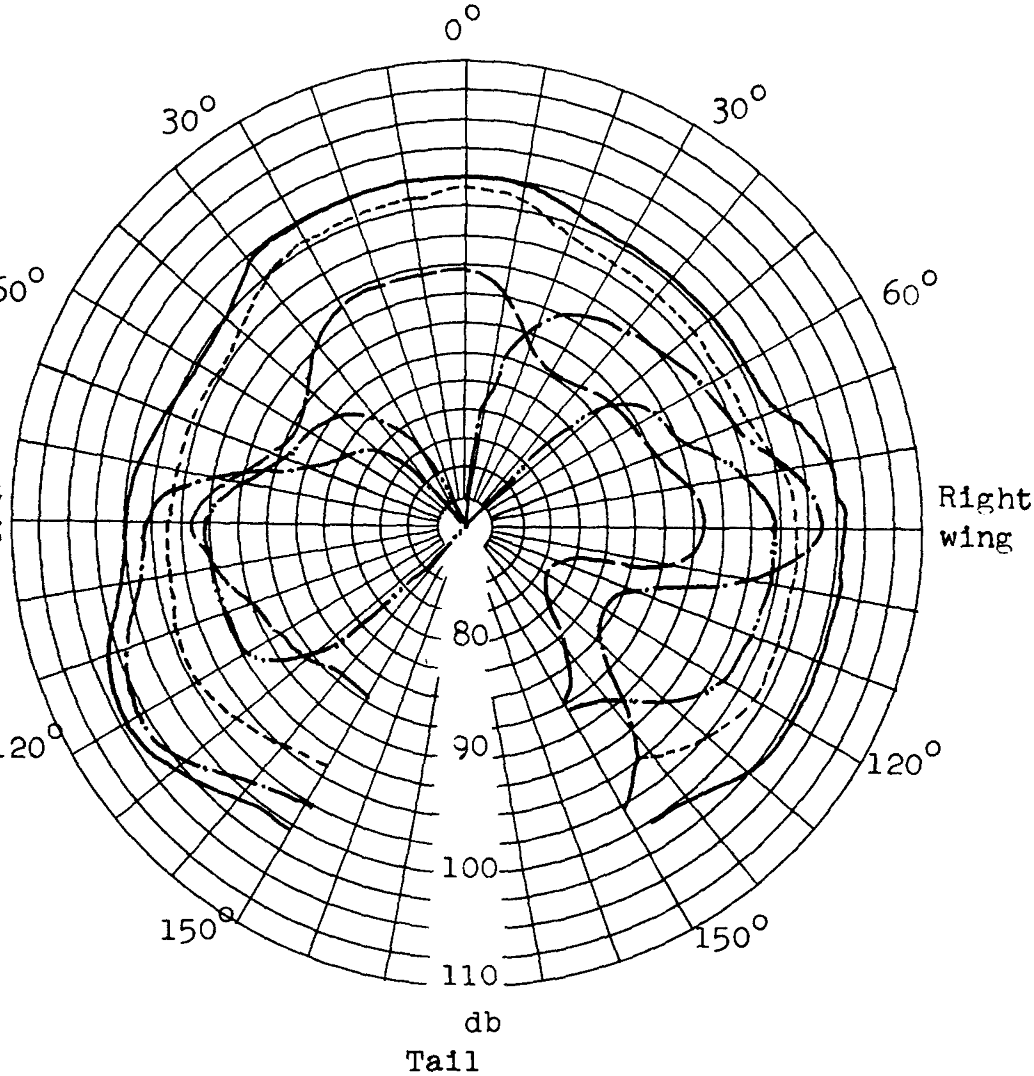
For example, according to Eq. (4.2), the overall power level is 152 db for engine delivering 500 horsepower under cruising conditions. According to Eq. (4.3), the overall power level is 160 db for an engine delivering 1000 horsepower at full load.

4.3 Total External Noise of Aircraft with Reciprocating Engines

According to Secs. 4.1 and 4.2, the overall noise level of a propeller increases by approximately 5.5 db per horsepower doubling (plus 2.7 db for each increase of 100 ft/sec in tip speed), whereas the overall noise level of an engine increases at approximately 3 db per horsepower doubling. It follows from these principles that the predominant noise source in a propeller-driven aircraft with very large engine power will be the propeller, but that engine noise will predominate when the power is low.

This expectation appears to be borne out in the results of a survey 3/ of take-off noise level of various airplanes ranging from 65 to 5800 horsepower. In this survey the microphone was located in the propeller plane at a distance of 500 ft from the center of the runway. At this microphone position the sound received from both engine and propeller has approximately the space-average value, so that directional effects may be neglected. It is found that the observed sound levels for aircraft with more than 150 horsepower agree with values predicted from the propeller chart, Fig. 4.1, to the accuracy of the chart. For airplanes of 150 horsepower and less, the overall noise levels exceed those predicted from the propeller chart, but are in approximate agreement with levels for engine noise as given by Eq. (4.2). There are, however, other take-off noise data 4/ for aircraft with less than 200 horsepower which are in agreement with propeller noise figures rather than with estimated noise figures. The reason for the discrepancy is not known.

A convenient approximate expression for the overall power level of various aircraft under take-off conditions



Engine — Average frequency
Fundamental — — — — — 97
Second harmonic 195
——— ——— ———

Propeller — Average frequency
Fundamental — — ——— — 65
Second harmonic——- - ——130
Third harmonic — — — — ——

Overall level ————

has been deduced from the data of Ref. (3). This relation is

$$\begin{pmatrix}\text{Overall power} \\ \text{level, take-off}\end{pmatrix} = 121 + 12 \log_{10} \begin{pmatrix}\text{Total take-off} \\ \text{horsepower of} \\ \text{aircraft}\end{pmatrix} . \qquad (4.4)$$

It happens that under the particular conditions found in take-off it is not necessary to consider propeller tip speed explicitly. Since the tip speed is not considered in Eq. (4.4), this relation cannot be applied to operating conditions differing materially from take-off.

The broken line drawn across the propeller noise chart, Fig. 4.1, divides the chart approximately into a region in which the propeller is the major noise source for an entire aircraft (upper right-hand portion) and a region in which the engine is the major noise source (lower left-hand portion). This line is constructed by computing, for various values of total horsepower, the tip speed at which the overall propeller noise power level equals the overall engine noise power level given by Eq. (4.2). A three-blade propeller with pointed tips is assumed. Operating data for small single-engine aircraft often fall in the region in which engine noise is important (lower left). All data used in deriving this dividing line represent average trends from which results for a particular aircraft may differ by as much as 5 db as regards either engine noise or propeller noise. Therefore, the line as drawn on the chart will not indicate accurately under what conditions propeller noise is dominant in a particular aircraft. Also, the results are averages for reciprocating engine aircraft as commercially produced up to 1952, and do not apply to specially constructed units in which noise-control measures are incorporated. It has been shown that overall aircraft noise can be reduced significantly by use of propellers with an increased number of blades and by use of exhaust mufflers. 5/

The preceding remarks on total aircraft noise refer to the space average of sound output. This concept is inherent in the definition of power level. In general, actual observation made in the propeller plane will agree approximately with space-average results. The directivity patterns

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Figure 4.5

Directional distribution of SPL for certain discrete-frequency components of airplane noise. Measurements 50 ft from hub; ground test at cruising power. Two-blade propeller; 1940 rpm; direct drive; 97 horsepower; blunt tips, speed 646 ft/sec. (From Fig. 27 a of Ref. 4).

of engine and propeller must be considered in predicting noise levels observed in other directions. A typical situation is illustrated in Fig. 4.5, which shows the variation with azimuth angle of measured overall sound level and of the levels of selected propeller and engine noise components for a particular small airplane. These results were obtained by operating the airplane at cruising power on the ground and by placing the microphone in various locations 50 ft from the propeller hub, and approximately at the hub level.

The tentative conclusions regarding external noise of reciprocating-engine aircraft are summarized below.

1. For large aircraft, the overall PWL for either cruising or take-off conditions is approximately equal to the overall PWL for propeller noise (Fig. 4.1).

2. For aircraft of 150 horsepower or less, it appears that the overall PWL under cruising or take-off conditions is approximately that given for the engine by Eq. (4.2).

3. The SPL in the propeller plane is approximately that which would be produced by a non-directional source having the stated overall PWL.

4. The overall PWL for various aircraft under take-off conditions is given approximately by Eq. (4.4).

5. The observed noise for positions directly ahead of, or directly behind, the aircraft is approximately the engine noise alone having a PWL given by Eq. (4.2) or (4.3).

6. The observed SPL for positions from 20° to 30° behind the propeller plane has the greatest preponderance of propeller components. An approximate indication of the overall SPL for this region may be obtained, for either cruise or take-off conditions, by adding 5 db to the value obtained by using the overall propeller PWL (Fig. 4.1) and proceeding as for a non-directional source.

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CHAPTER 5

AIRCRAFT JET AND ROCKET ENGINES

5.1 Introduction

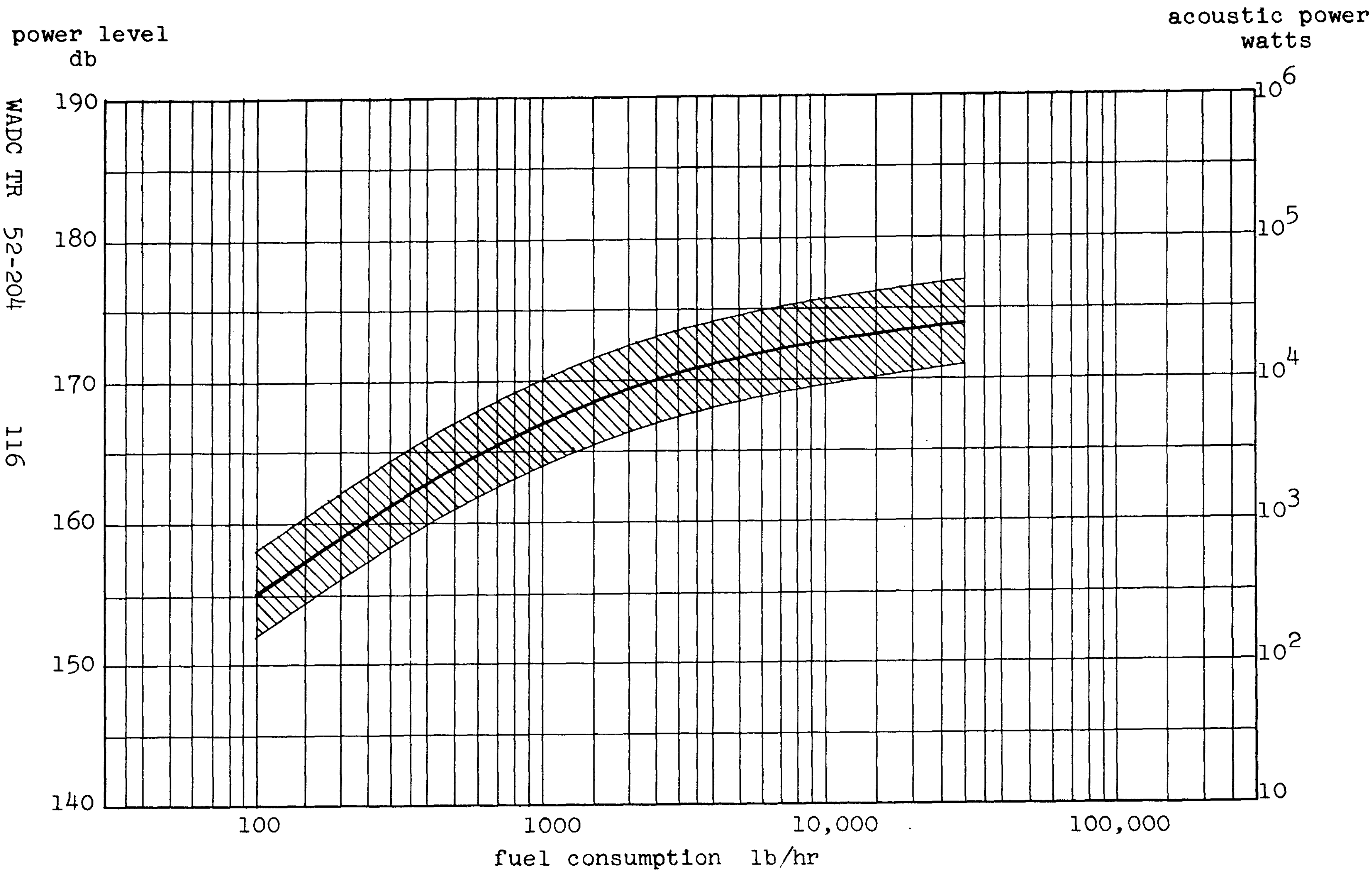
No generally accepted theory exists at the present time which will explain the production of noise by burning jets and rocket engines. Observations have indicated that to a first approximation the noise energy produced by an aircraft jet or rocket engine is proportional to the kinetic energy of the jet stream. Consequently, in the absence of a well established theory on the generation of noise from burning jets and rocket engines, the sound energy output of these engines has been related empirically to parameters which are a function of this jet stream kinetic energy. The specific cases of ram jets, turbo jets and rockets will be discussed in detail below.

The empirical relationships, derived on the basis of many measurements, enable one to estimate the sound power output of an aircraft jet or rocket whose engine operating conditions are known. In addition, information presented in the following sections will enable one to estimate the spectral distribution of the sound energy from any class of aircraft engine discussed and, in some cases, the directionality characteristics of these engines.

5.2 Ram Jet Engines

Power Levels. Many measurements have been made on ram jet engines of all sizes, over a wide range of fuel consumption and fuel-air ratio. These measurements have been made on ram jets operating inside a closed test cell, in a semi-closed test cell, and in an open field. In all cases, the data taken in the form of sound pressure levels have been converted to power levels in decibels re 1.0×10^{-13} watts. In instances where more than one measurement was made on the same engine for the same rate of fuel consumption, the resultant power levels have been averaged.

A plot of these power levels vs fuel consumption of the ram jet engine in lb/hr results in the curve shown in Fig. 5.1. The heavy solid line is an estimated best fit of the data, and the cross-hatched region is the present range of uncertainty (± 3 db).



power level
db
190
180
170
160
150
140
acoustic power
watts
10^6
10^5
10^4
10^3
10^2
10
100
1000
10,000
100,000
fuel consumption lb/hr

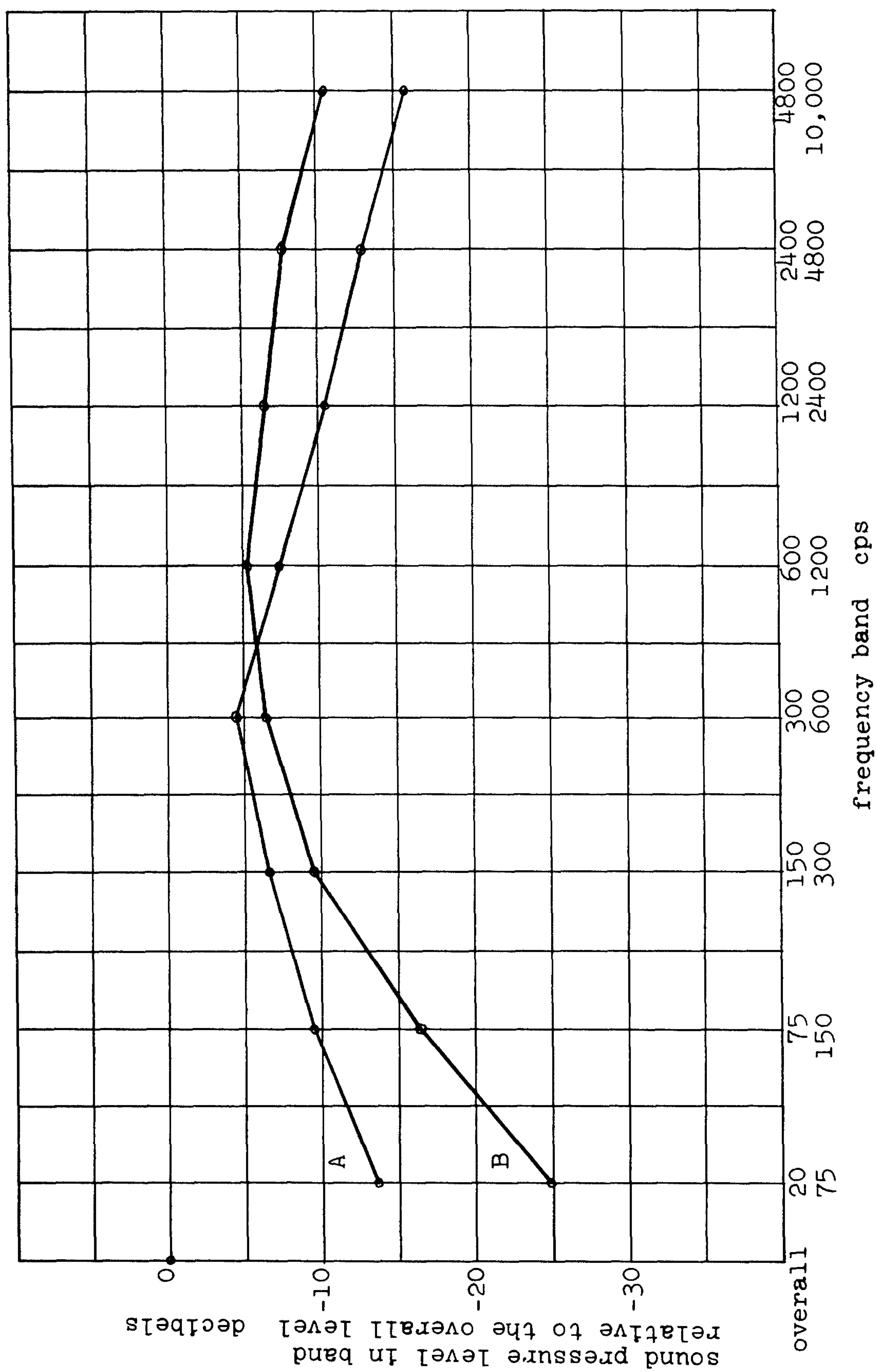
Examination of this power level curve reveals that in the range of low fuel consumption, the power level increases linearly at about 4 db per doubling of fuel consumption. However, as the fuel consumption is increased, the power level curve flattens off until in the higher range of fuel consumption the increase in power level is only about 1 db per doubling of fuel consumption. It is interesting to note that the sound power output of a ram jet engine for high values of fuel flow is in excess of one kilowatt, as can be seen from the right hand ordinate in Fig. 5.1 which gives acoustic power in watts.

Frequency Spectrum of Ram Jets. From the measurements made on ram jets, it has been found that in general the frequency spectrum follows a particular pattern. The small ram jets (those having combustion chamber diameters less than 9 in.) produced lower sound pressure levels in the lower frequency range than the large ram jets (those having combustion chamber diameters greater than 16 in.). Furthermore, in the higher frequency range the small ram jets produced higher sound pressure levels than the larger ram jets. The typical frequency spectrum for each range of ram jet size, as calculated from the averages of the measured spectra of the several ram jets measured, are shown in Fig. 5.2. It is seen from these curves that on the average the peak in the spectrum for the large ram jets occurs in the 300-600 cps band, and for the small ram jets, in the 600-1200 cps band.

It should be emphasized that these frequency spectra represent average curves. In general. the noise spectrum of any one particular ram jet is different from that of another design of ram jet. Apparently the diameter, length, and constructional details influence the noise spectrum in some way. However, enough data are not available to make an analysis of the influence of these variables upon the noise output of a ram jet engine. In addition to these variables, the directivity pattern of the ram jet noise will influence the frequency spectrum to a certain extent. However, these typical spectra are representative of the noise output of ram jet engines operating under smooth-burning conditions.

Figure 5.1

Power level of ram jet engines and turbo jet engines with afterburners as a function of fuel consumption in lb/hr. The ordinate scale on the right gives the total acoustic power radiated by the engine.


sound pressure level in band
relative to the overall level decibels
0
-10
-20
-30
overall
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps
A
B

Estimation of Levels. With the aid of the information contained in Figs. 5.1 and 5.2, it is possible to estimate the noise spectrum from a ram jet engine operating under smooth-burning conditions. The way in which this is done is illustrated by the following example. It is desired to determine the noise spectrum 500 ft from a ram jet having a combustion chamber of 23 in. in diameter and consuming fuel at the rate of 5000 lb/hr. From Fig. 5.1 it is noted that the power level for this rate of fuel consumption is 171 db. If hemispherical radiation is assumed, the sound pressure levels at 500 ft can be determined by the following formula

$$SPL = PWL - 10 \log_{10} 2 \pi (500)^2$$

This calculation results in an overall sound pressure level of 109 db at 500 ft.

From Fig. 5.2 we note that for ram jets having a combustion chamber diameter of 16 in. or greater, the level in the 20-75 cps band is 13.5 db below the overall level. Consequently, the SPL in the 20-75 cps band at 500 ft is 95.5 db. In a similar manner the levels in the remaining octave bands have been calculated and are tabulated below:

20-75 cps	95.5 db
75-150	99.5
150-300	102.0
300-600	104.5
600-1200	101.5
1200-2400	98.5
2400-4800	96.0
4800-10000	93.0
overall	109.0

5.3 Turbo Jet Engines

In this section, information will be given on the noise characteristics of turbo jet engines. There are many indications that at a given operating condition there may be more

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Figure 5.2

Typical spectra for ram jet engines and turbo jet engines with afterburners under smooth-burning conditions. Curve A applies for engines having combustion chamber diameters of 16 in. or greater; Curve B is for engines having combustion chamber diameters of less than 9 in.

than one mechanism responsible for the production of noise in a turbo jet engine. In addition, the mechanisms at one operating condition may differ considerably from those at another operating condition. Consequently, we should expect that the noise characteristics of turbo jet engines may depend on one set of parameters in one range of operating conditions, and that they may depend upon other parameters at different operating conditions.

At present, two noise-producing mechanisms appear to be important:

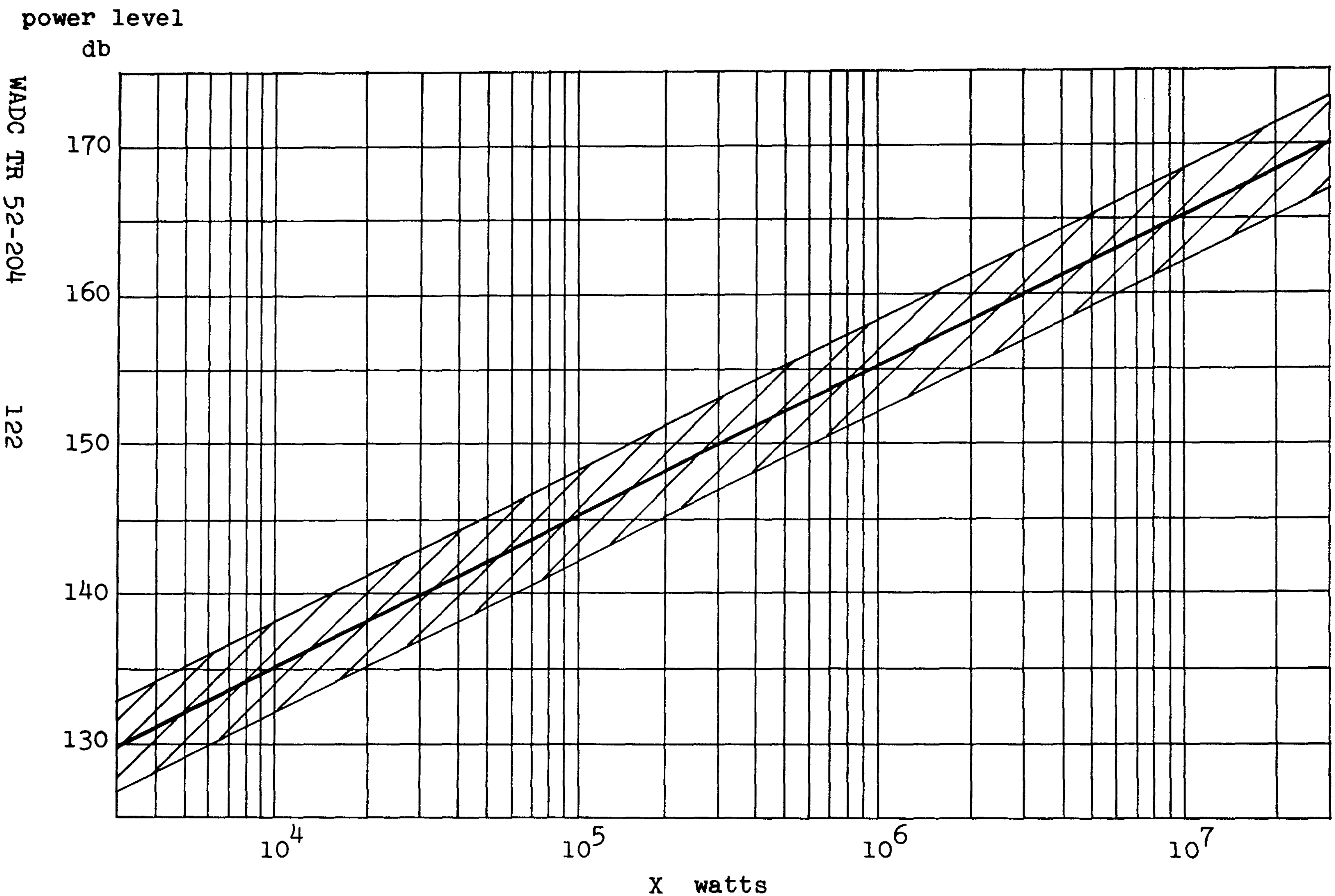
1. aerodynamic turbulence in the jet stream
2. rotation of the turbine and compressor blades.

The relative importance of the two mechanisms listed above depends upon the operating conditions, the angular position of the observer relative to the engine, and the frequency. For example, at military power, mechanism (1) is generally more important at all frequencies and at all angles. But at idling power, mechanism (1) is more important only for angles toward the exhaust of the engine for all frequencies, while for angles toward the intake of the engine, (2) is more important for high frequencies, and (1) is more important for low frequencies.

In general, the addition of an afterburner to a turbo jet considerably changes the noise source characteristics. Engines with afterburners consume about three times the fuel, and the temperature of the exhaust is correspondingly greater. These conditions correspond roughly with the conditions of ram jet operation, and measurements have shown that the ram jet design curves, given in Sec. 5.2, may be used for turbo jets with afterburners. Consequently, this section will contain information only on turbo jets without afterburners.

Power Levels of Turbo Jets. The power level of turbo jet engines for noise mechanism (1) is presented in Fig. 5.3. The power level has been correlated to a parameter X, where

$$X = W \left[1 + \left(\frac{W T_o d_o}{W_o T d}\right)^3\right] \quad \text{watts} \qquad (5.1)$$


power level
db
170
160
150
140
130
10^4
10^5
10^6
10^7
X watts

and where

$$W = [5.70] \frac{S^3 (T - T_o)^3}{T} \left[\frac{A}{286} \right] P \quad \text{watts} \qquad (5.2a)$$

or

$$W = [0.90] \frac{v^3}{T} \left[\frac{A}{286} \right] P \quad \text{watts} \qquad (5.2b)$$

$W_o = 2.03 \times 10^6$ watts

$T_o = 1450^\circ$ Rankine

$d_o = 19$ inches

d is the jet nozzle diameter in inches

S is the ratio of static thrust in pounds to fuel consumption in lb/hr

T is the tail-cone temperature, in degrees Rankine

T_o is the intake temperature, in degrees Rankine

A is the cross-sectional area of the tail cone in sq in.

P is the exhaust pressure in atmospheres

v is the average jet velocity in ft/sec.

An example of the use of Fig. 5.3 is included in Sec. 12.14.

The noise due to mechanism (2) may be determined from the information on compressor noise given in Chapter 6. The parameters necessary to determine the noise are the horsepower driving the compressor, the rpm and the number and diameter of the blades.

Of the two mechanisms, (1) is the greater source of noise for X larger than 5×10^6 watts. From the noise control point of view, therefore, we have considered in the paragraphs below

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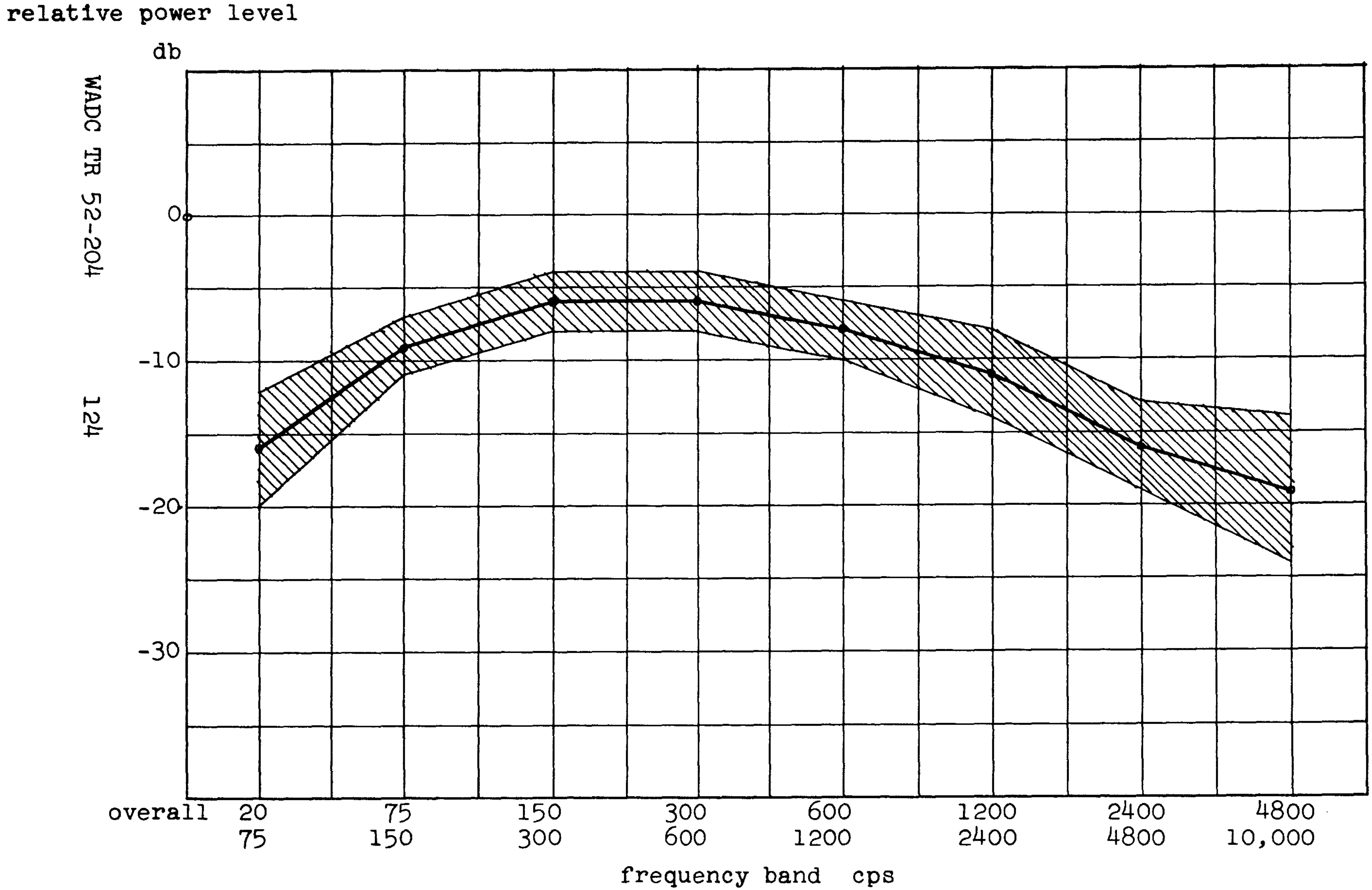
Figure 5.3

Power level of turbo jet engines without afterburners as a function of the parameter X which is defined in Eq. (5.1). This curve applies to the noise generated by turbulence in the jet stream. Seventy percent of the measured points fall within the shaded region (± 3 db) and ninety percent fall within a band ± 5 db wide.


relative power level
db
0
-10
-20
-30
overall
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

on directionality and spectra only the properties associated with mechanism (1).

Directionality of Turbo Jets. For noise mechanism (1) and for X greater than 5×10^6 watts, the maximum of the radiation from a turbo jet engine occurs at an angle between 30° and 60° from the jet exhaust. 1,2,3,4/ The directivity index at the angle of the maximum is 8 db (±2 db).

The power radiated between 0° and 90° from the exhaust jet is 13 db (±2 db) higher than the power radiated between 90° and 180°. In a test cell, therefore, one would expect the sound power in the exhaust section to be 13 db (±2 db) higher than in the intake section if the exhaust and intake sections were completely isolated. In a given test cell, the sound isolation afforded by the wall separating the two sections can be estimated, and the difference in sound level between the two sections can then be obtained.

Spectra of Turbo Jets. The spectrum of turbo jet engines can be obtained from Fig. 5.4. The spectrum is valid for noise mechanism (1), for X greater than 5×10^6 watts, and for engines whose tail cones are approximately 19 in. in diameter. The spectrum should be shifted one octave lower per doubling of the tail cone diameter.

5.4 Rockets

A limited number of measurements on noise generated by rocket engines indicates that the sound power output of a rocket can be correlated with the rocket's fuel consumption, in a manner similar to that discussed above in Sec. 5.2 on ram jet engines. More specifically, it has been found that

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Figure 5.4

Typical spectrum for turbo jet engines without after-burners. This curve applies only to noise generated by turbulence in the jet stream and for $X > 5 \times 10^6$ watts. Eighty percent of the measured points are within the shaded region and an additional five percent fall within 3 db of the dotted boundaries. The measurements were made on an engine with a 19 in. tail cone diameter. Shift the spectrum down one octave for each doubling of tail cone diameter.

the power level of a rocket engine can be related to the fuel consumption by the following expression

$$PWL = 149 + 10 \log_{10} (F)^{1/2} \text{ db} \tag{5.3}$$

where F is the fuel consumption (oxidizer plus fuel) in lb/hr. The measurements from which the above relationship was derived were made on rockets operating in open-ended test cells and exhausting into open fields. Several measurements were made in the vicinity of the rocket exhaust and the results of these measurements were averaged to determine the power level.

From these measurements it was also possible to determine an average rocket noise spectrum. This spectrum agrees in general with an average of the two typical ram jet spectra shown in Fig. 5.2.

The directional pattern of noise radiated from the exhaust of a rocket engine varies with the rocket diameter, fuel consumption, thrust, and with frequency. In general the maximum overall sound power occurs at an angle of approximately 45°. The sound power radiated in any one octave band, however, may be a maximum anywhere from 10° to 90° from the rocket exhaust. Variations in sound pressure level over this angular range sometime exceed 15 db.

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CHAPTER 6

FLUID FLOW DEVICES

6.1 Introduction

This chapter deals with the production of noise by devices which control or produce fluid flow. The basic principles which are evolved are applicable either to liquid-flow or to gas-flow. The specific discussion deals with air-flow devices, for which experimental data are available.

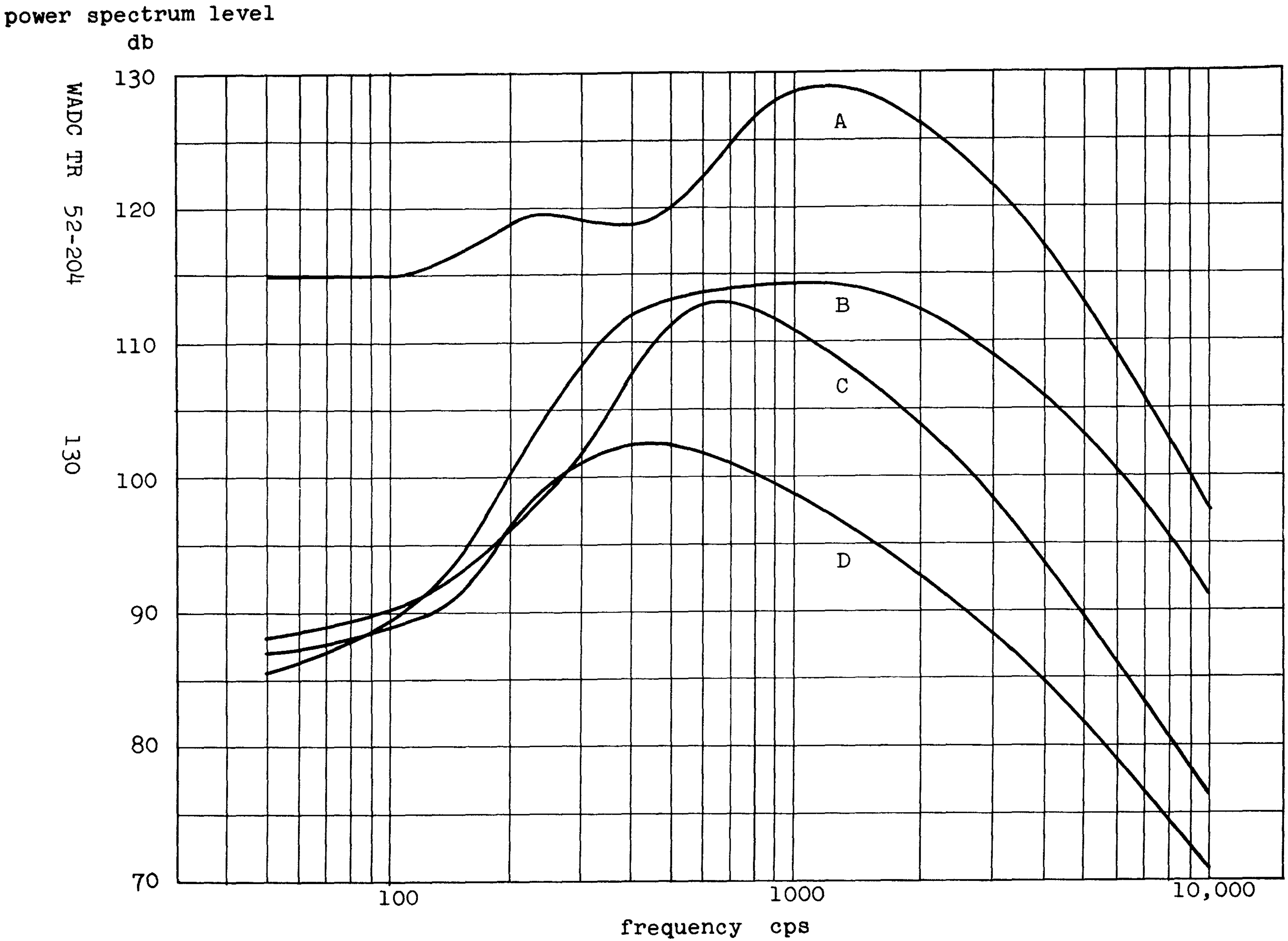
The parameters which occur in the discussion of fluid-flow noise include speed of flow, volume rate of flow, pressure drop, and fluid density. In the case of turbine compressors, all of the parameters which were used in the description of aircraft propeller noise (Chapter 4) apply. It is often possible, however, to give a simplified description of a fluid-flow noise source in which the important parameter is power input or power dissipation. The power is a single quantity which conveniently represents a combined effect of several parameters.

6.2 Wind Tunnels

In aeronautical test facilities, wind tunnels are one of the major noise sources. The wind tunnels considered in this section have no provisions for fuel combustion. The contribution of combustion to the noise is described in Sec. 5.3.

The noisiest part of a wind tunnel (supersonic or transonic) is generally the diffuser. Moreover, the noise seems to originate in the vicinity of the normal shock, which usually takes place in the diffuser. As a result of this observation, it has often been erroneously believed that the normal shock is the main source of noise. It appears, however, that the noise generated inside the diffuser is due to the turbulent character of the fluid flow. The noise level, then, should increase with the turbulent losses in the system. The turbulent losses increase linearly with the density of the fluid and with the cube of the average flow velocity. Measurements indicate that the acoustic energy is indeed proportional to the density of the fluid and to the cube of the flow velocity.

In this section an approximate theoretical analysis is presented to systematize the results of several experimental observations of noise produced by supersonic wind tunnels. The experimental results, obtained from several different tunnels, are for flow velocities in the range of Mach Numbers from 1.2 to 5.


power spectrum level
db
130
120
110
100
90
80
70
100
1000
10,000
frequency cps
A
B
C
D

Some quantitative uncertainty is found in the results of the analysis which follows. Further measurements are necessary for more accurate results. The present uncertainty must be interpreted in the form of engineering tolerances when noise control structures are designed for diffuser installations.

Available Data

(1) Methods of Measurements

The procedure used in the field to measure the noise from a diffuser consists of bringing a microphone as close as possible to the tunnel on the outside. Then, by moving the microphone along the length of the diffuser, the position of maximum noise level is located. This usually corresponds to the position of the shock. To determine the spectrum of the noise, levels in eight octave bands are measured separately.

(2) Calculation of Noise Inside Tunnel

An estimate of the transmission loss through the walls of the tunnel is now made. By adding this transmission loss to the spectrum measured outside the diffuser, the spectrum of the noise inside the tunnel can be approximately evaluated. The error involved in this method of calculation is of the order of 5 decibels, which is not excessive on an absolute basis, since the levels usually encountered range from 100 to 150 decibels. Also, this amount of inaccuracy is practically unavoidable, owing to the inherent difficulties in making measurements inside a diffuser. Practically no data exist on direct noise measurements inside diffusers. Figure 6.1 shows some typical measurements on tunnels.

Figure 6.1

Summary of noise measurements on supersonic wind tunnels. The curve presents data taken at

A.	Daingerfield	Mach 2.5
B.	M.I.T.	2.5
C.	NOL (18 x 18 cm)	3.1
D.	NOL (40 x 40 cm)	2.9

The SPL was measured outside the tunnel in the vicinity of the shock and by estimating the TL of the walls, the SPL inside the tunnel was obtained. These octave band measurements were converted to power levels by correcting for the cross-sectional area of the tunnel at the point of measurement. Finally, the spectrum level plotted here was obtained by the method shown in Sec. 2.4.

Nature of Noise Spectrum. A critical examination of the diffuser of a wind tunnel reveals that the level of the noise generated there varies in an inverse manner to the efficiency of the diffuser. The efficiency of the diffuser should be considered in its broad sense, i.e., as an indication of the ability of the diffuser to allow for the reduction of the flow velocity with a minimum of losses.

In general it can be stated that the expected sound level increases with the losses. Furthermore, it is observed that the sound level is highest in the vicinity of the shock in the diffuser of a wind tunnel. Since there exists a large pressure differential across the shock and since the position of the shock wave is fluctuating about a mean position, it might be thought that the nature of the noise is really due to a fluctuating force concentrated in the vicinity of the shock and pounding on the walls of the tunnel.

Consideration of this hypothesis leads one to expect that the spectrum of a noise generated in this fashion would exhibit a prominent peak at the frequency of vibration of the shock, and several other peaks corresponding to the mechanical resonances of the walls. However, this is completely contradicted by the shape of the noise spectra found in the field measurements.

When it is remembered that losses in the diffuser are intimately connected with the appearance of turbulence, the statistical nature of the noise 1/ associated with the diffuser is expected to be related to the statistical distribution of velocities in turbulent flow.

It is proper at this point to summarize briefly the current theories of turbulent flow, to help in the interpretation of the data collected in the field measurements and also to lay a firm basis for any estimates to be made.

In a series of papers of fundamental importance to the theory of turbulence, Taylor 2/ established that a length ℓ, "the mixing length" can be derived. The mixing length characterizes the turbulent flow and is of basic meaning to the theory, just as the mean free path concept is an essential quality in the kinetic theory of gases. The mixing length ℓ depends on the dynamic condition of the fluid and is to a first approximation independent of the physical constants of the medium (such as thermal conductivity, etc.).

In a subsequent investigation Taylor 3/ showed that the spectrum of an isotropic * turbulent flow is the Fourier transform of the correlation function of the velocity at any two points chosen in the turbulent field. Taylor's theories have been verified on turbulent field created by interposing a grid-like structure in a stream of moving air. The predicted value for the mixing length ℓ of such an arrangement (the side of a mesh of the grid) was found to agree well with the measurements. Furthermore, Taylor pointed out that whenever the frequency variable ν is replaced by ν/U, U being the flow velocity, then for a turbulent field of the same mixing length, the spectrum referred to ν/U is independent of the flow velocity. The experimental verification of this was reported by Simmons and Salter. 4,5/

The conditions existing inside the diffuser are different from the idealized, isotropic turbulence considered by Taylor. Inside a conduit the turbulent field departs appreciably from isotropicity, and as a result the mixing length ℓ becomes a function of position. 6/ The conclusions found in the isotropic case are not directly applicable here, but it is safe to state that the spectrum of turbulence, for the same conduit, will be independent of the flow velocity provided the frequency ν is transformed to ν/U.

Another proposition which seems reasonable is that for geometrically similar conduits, the spectrum when made a function of the argument $\nu L/U$, where L is a characteristic length of the conduit, should be invariant. The investigations of Reichardt 7/, who studied the fluctuations of the velocity in conduits, and especially of Motzfeld 8/, were along these lines. The latter determined experimentally the spectrum distribution of the turbulence inside conduits of rectangular section, and was able to verify that the spectrum referred to $\nu L/U$ was indeed variant.

Following Motzfeld, all the available data on diffusers have been plotted on a single graph, after rendering the frequency dimensionless through the transformation $\nu L/U$. The

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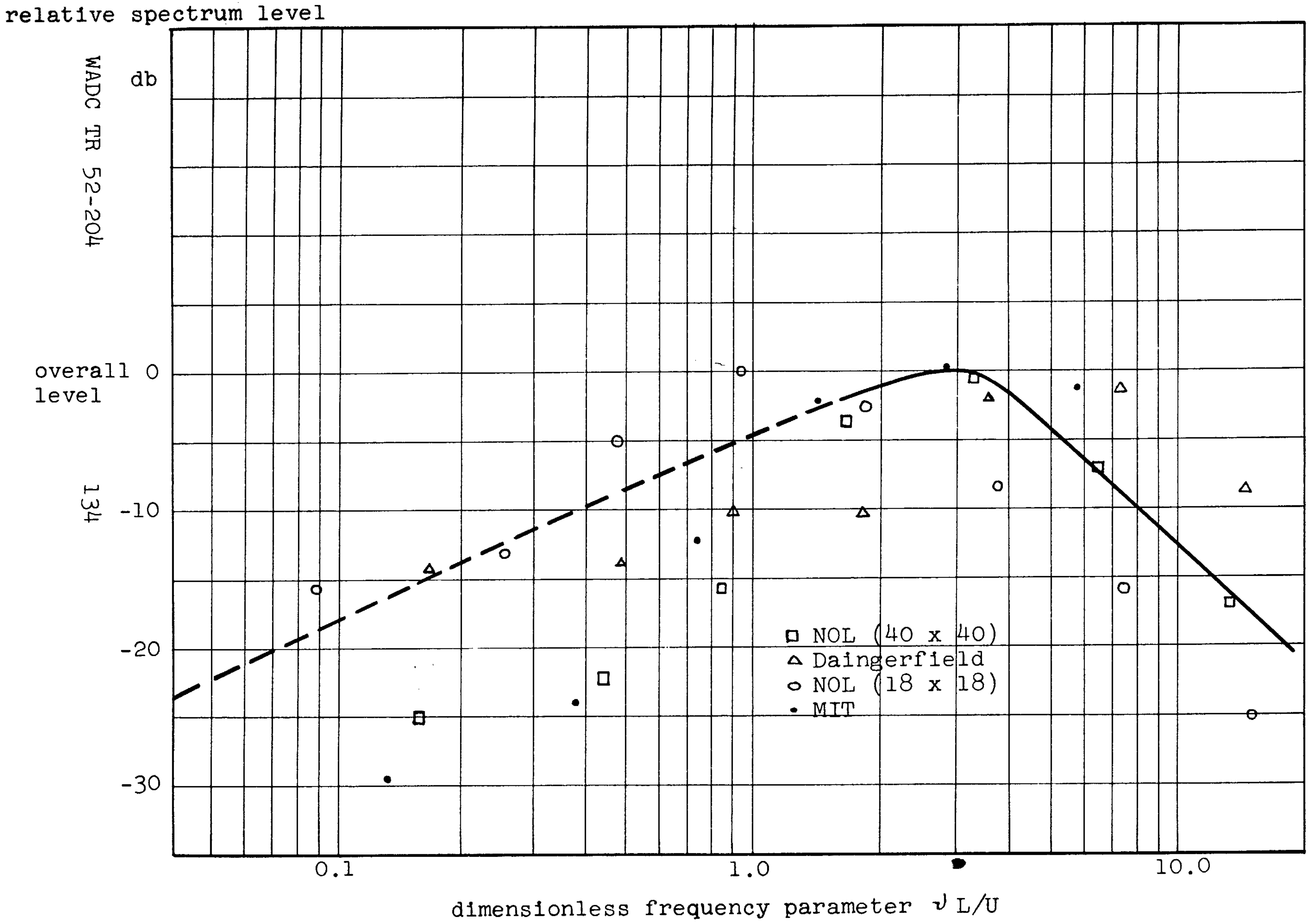
* "In isotropic turbulence the average value of any function of the velocity components, defined in relation to a given set of axes, is unaltered if the axes of reference are rotated in any manner." See Ref. (2).


relative spectrum level
db
overall 0
level
-10
-20
-30
0.1
1.0
10.0
dimensionless frequency parameter ν L/U
NOL (40 x 40)
Daingerfield
NOL (18 x 18)
MIT

characteristic length L was taken as the geometrical mean of the lengths of the sides of the cross-section of the diffuser in the vicinity of the shock. The flow velocity U is that which occurs just downstream from the shock. It might be supposed that L is related in some manner to the mixing length inside the conduit. This functional dependence varies for different geometries of the conduit but is expected to be the same for similar sections. The replotted data are shown in Fig. 6.2, which also includes Motzfeld's experimental curve. The data are normalized in height, to have the same maximum. Upon inspection of Fig. 6.2 it is seen that the measured data fit Motzfeld's curve reasonably well considering the uncertainties in the method of measurement. The trend of the points indicates that the nature of the noise can be attributed to the turbulent flow inside the diffuser.

A better fit of the data to Motzfeld's curve can be obtained if the value of L is multiplied by a weighting coefficient. There are at least two ways of interpreting this fact: (1) the diffusers are not geometrically similar, so that the functional relation $L(\ell)$ is different for each diffuser; (2) the decay of turbulence down the diffuser makes it difficult to sample the noise at a place where all diffusers will exhibit similar spectra.

However, in spite of these slight modifications in the physical conditions, it can be stated that the spectrum of the noise is that of the turbulence set up in the diffuser and Motzfeld's curve reduced to dimensionless frequencies is a satisfactory approximation for the predicted spectrum.

Figures 6.3 (a) and (b) show a comparison of the individual measured curves and Motzfeld's curve as calculated for each particular case. The comparison is seen to be fairly good.

Evaluation of Power Level. A method of estimating the acoustic power level in the vicinity of a given tunnel as a function of the flow parameters inside the tunnel is required.

On the basis of dimensional analysis 8/, the turbulent losses for geometrically similar conduits seem to be proportional

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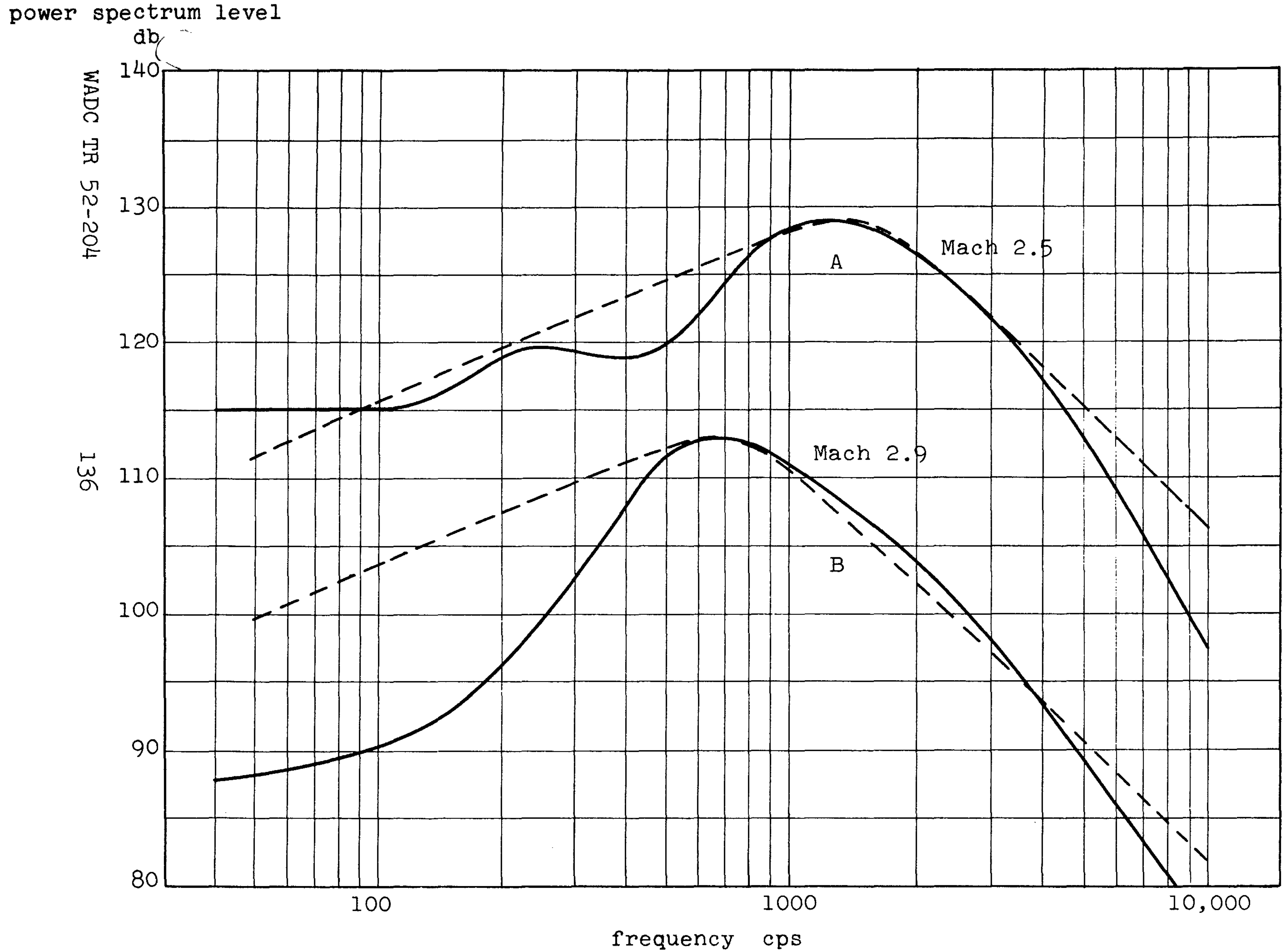
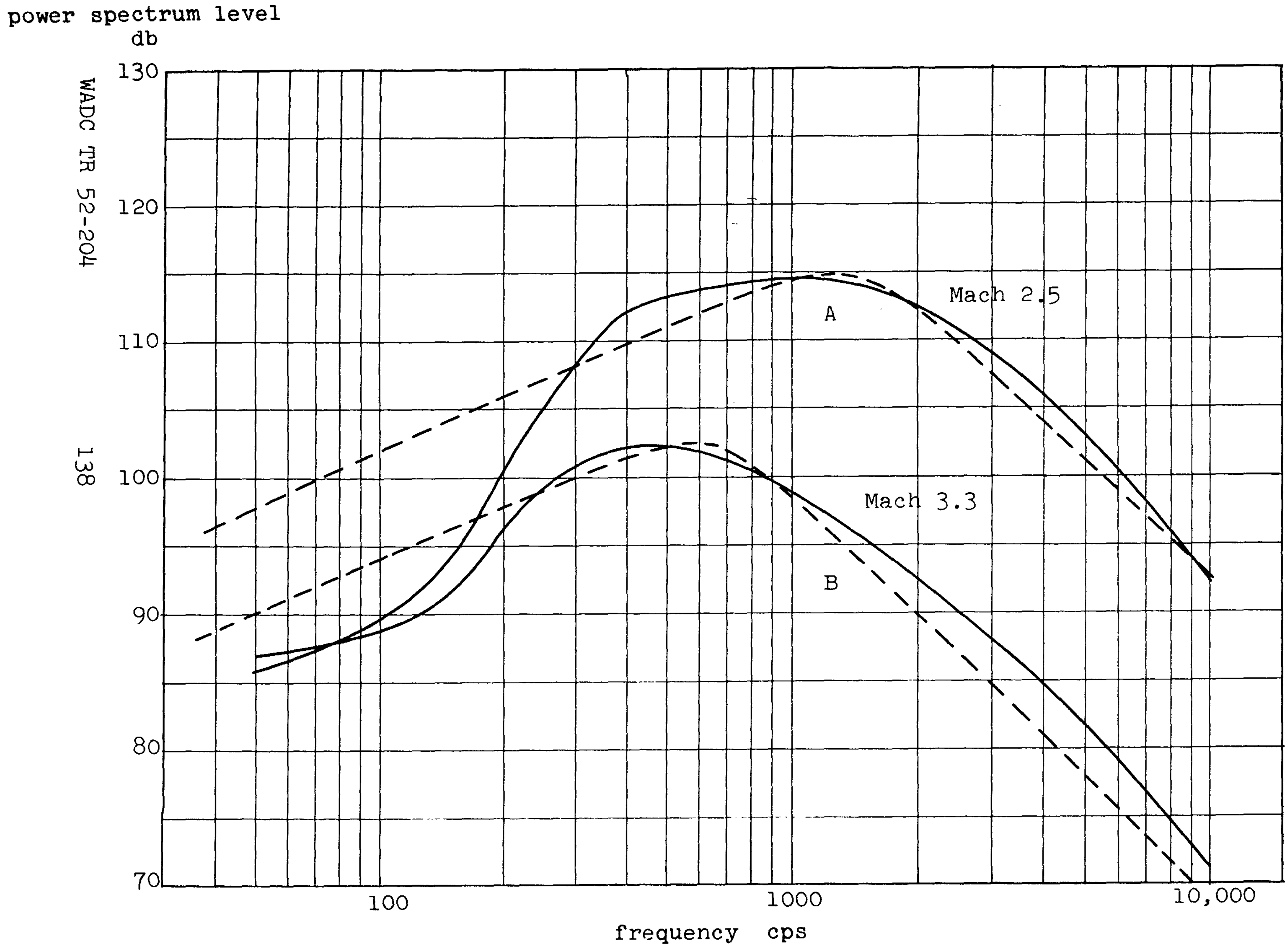
Figure 6.2

Spectrum of noise reduced to the dimensionless parameter L/U compared with the data of Motzfeld.


power spectrum level
db
140
130
120
110
100
90
80
Mach 2.5
A
Mach 2.9
B
100
1000
10,000
frequency cps



power spectrum level
db
130
120
110
100
90
80
70
100
1000
10,000
frequency cps
Mach 2.5
A
Mach 3.3
B

to ρU^3 where ρ is the density and U is the average velocity of flow. This quantity is related to the kinetic energy of flow, since

$$\rho U^3 = (\rho \times U) \quad U^2$$
$$= (\text{mass flow/unit time}) \times U^2.$$

If it can be shown that the acoustic power is directly proportional to the turbulent losses, it should be possible to evaluate an efficiency of conversion of mechanical to acoustic power. For that purpose, acoustic power radiated from several wind tunnels as a function of the kinetic energy of flow, referred to a region immediately behind the normal shock in the diffuser, was measured. The kinetic energy is referred to that region since the flow is totally turbulent there.

Figure 6.4 shows the result of the survey. It indicates that an efficiency of conversion can be defined over a wide range of operating conditions. The figure also indicates that the representative points of the several tunnels can be grouped in two classes (each corresponding to a straight line). These classes refer to tunnels with or without a second throat in the diffuser. It then becomes a simple matter to estimate the character of the noise in the vicinity of the tunnel.

Procedure for Design

1. Identify the method of operation of the tunnel (with or without a second throat in the diffuser).

2. Given the stagnation conditions (pressure, temperature), the nominal test-section area, and the Mach number, evaluate the flow conditions immediately behind the shock in the diffuser.

3. Evaluate the kinetic energy of flow behind the shock.

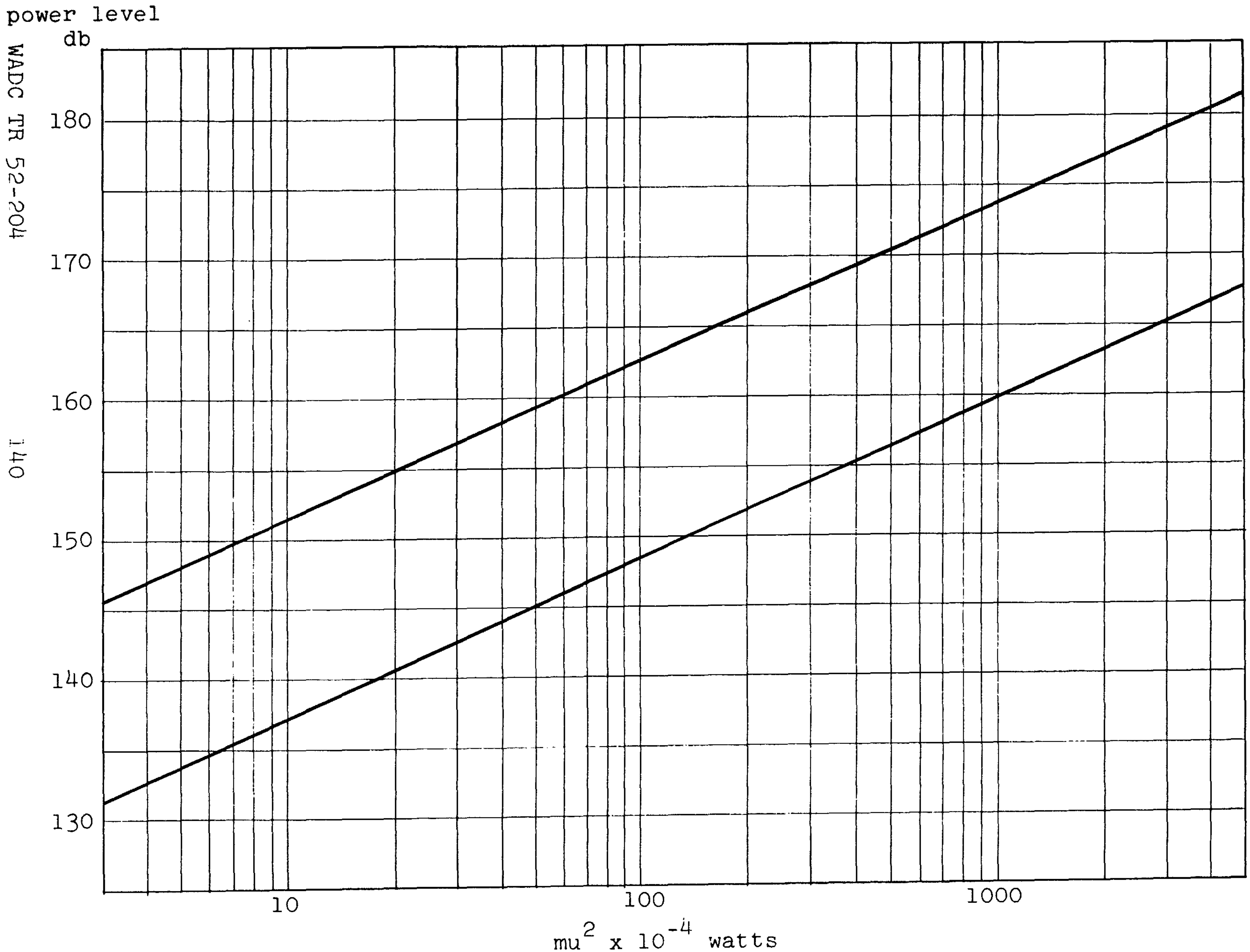
Figure 6.3

(a) Individual comparison of data with the Motzfeld curve

A	Daingerfield	Mach 2.5
B	NOL (40 x 40 cm)	2.9

(b) Individual comparison of data with the Motzfeld curve

A	M.I.T.	Mach 2.5
B	NOL (18 x 18 cm)	3.3


power level
db
180
170
160
150
140
130
10
100
1000
$mu^2 \times 10^{-4}$ watts

4. From Fig. 6.4, determine the expected acoustic power level (adding 10-15 db if the diffuser has no second throat).

5. The spectrum of the noise is approximately the Motzfeld curve, the maximum point of which is located for a dimensionless frequency $\nu L/U$ of about 3.0.

6.3 Compressors

Centrifugal Compressors. Noise levels have been measured for five different centrifugal compressors of various speeds and driving motor horsepower of three different manufacturers. The results of these measurements are shown as circles in Fig. 6.5 in a plot of power level vs. horsepower for various blade tip speeds.

Because of the acoustic similarity between the compressor blades and aircraft propellers, the theory and design procedures given in Chapter 4 are applied here without change. The PWL increases 5.5 db for every doubling of horsepower. The spacing of the lines of constant blade tip speed in Fig. 6.5 is obtained from Fig. 4.1.

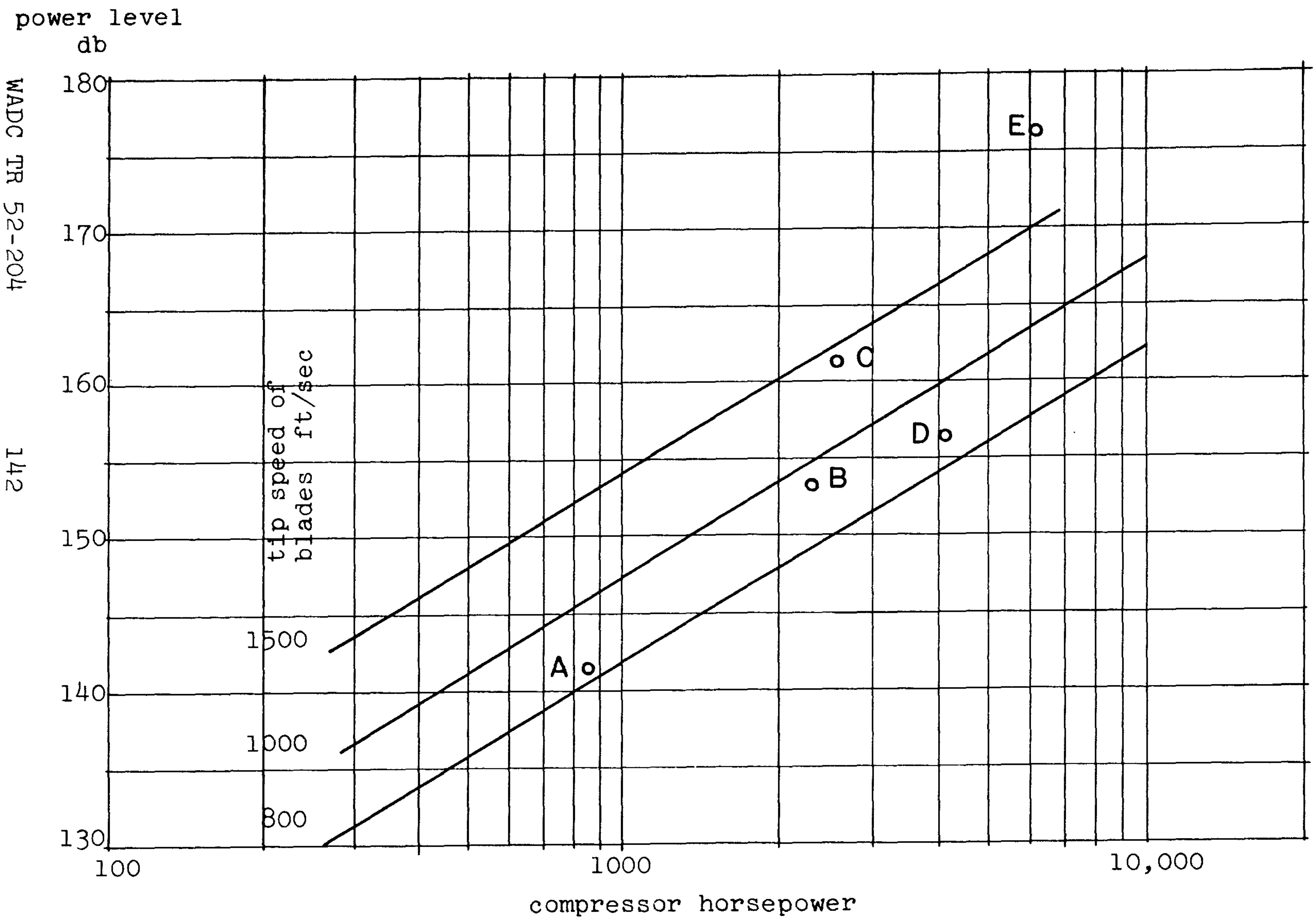
The agreement of the measured points is seen to be excellent in four cases but the power level for the Ingersoll-Rand compressor in the MIT supersonic tunnel is 10 db greater than the chart would indicate. Such a deviation is understandable when the measuring difficulties involved are understood. The SPL was measured outside the intake or exhaust piping and the TL through the piping was estimated (see Sec. 11.5). Then the effective radiating area A must be known in order to compute PWL from the relation

$$\mathrm{PWL} = \mathrm{SPL} + 10 \log A.$$

In this case the associated piping is radiating sound because it is closely coupled to the compressor proper and so the area A is uncertain.

Figure 6.4

Relation of acoustic power level and kinetic energy of flow. The upper line refers to diffusers without a second throat; the lower curve refers to diffusers with a second throat.


power level
db
180
170
160
150
140
130
100
1000
10,000
compressor horsepower
tip speed of
blades ft/sec
1500
1000
800
A
B
C
D
E

The noise spectra are shown in Fig. 6.6 for the five compressors measured. It will be seen that the spectra vary considerably, particularly in the low frequency bands where there is a spread of 20 db. The noise in these low frequency bands is believed to be motor and gear noise; hence the spectra shown are valid only for the frequencies near the peak. Despite this variation in the spectra, the total acoustic power output can be found reasonably accurately because most of the noise is concentrated into one or two bands.

Axial-flow Compressors. For purposes of noise control design in connection with compressors, it is necessary to have quantitative information on the magnitude of the total noise radiated as a function of frequency over the audible range. At the present time, there is not available definitive information on noise from axial-flow compressors, but some measurements have been made on a few types. The designs of compressors can vary over wide limits of delivered horsepower, shaft speed, number of stages, number of blades, air capacity, and many other variables which may influence the noise output. Therefore, it is not a simple matter to take sound level readings on one machine and derive the expected acoustic performance of a compressor of different characteristics. This problem is particularly difficult when one tries to extrapolate the performance to a very much higher power than that represented in the measured data.

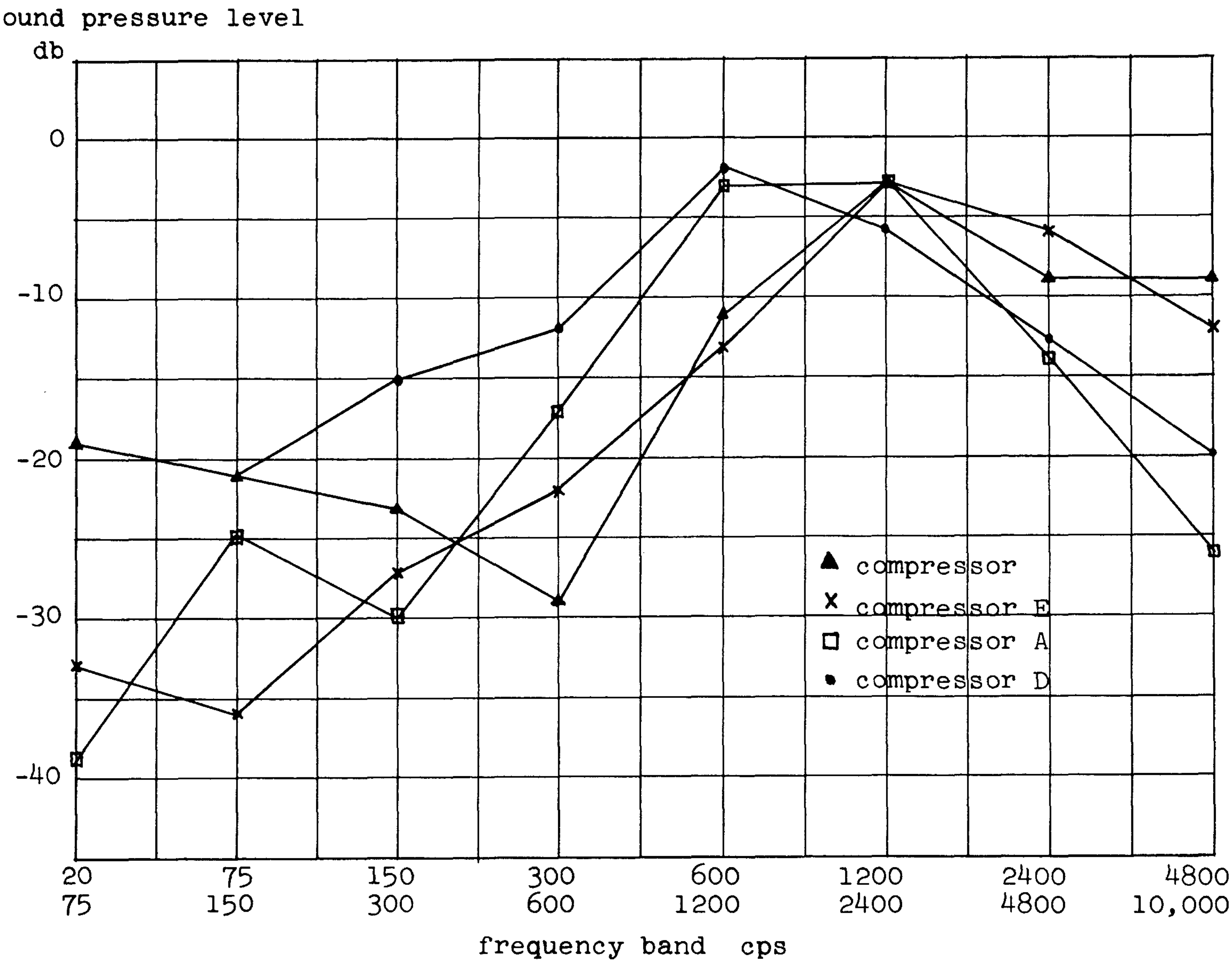
Nevertheless, it is necessary to obtain some estimate of the magnitudes involved, even though the estimated values may have wide engineering tolerances. A method of estimation is presented here in a form suitable for extrapolating to new designs, at least within certain limits. This empirical design procedure is based on measured noise spectra from several axial-flow compressors in the range of 300 to 2500 horsepower, and

---

Figure 6.5

Power level of noise inside the intake or exhaust piping of centrifugal compressors as a function of the compressor horsepower and the blade tip speed. The indicated data points are for the following compressors:

A	Worthington Pump	790 ft/sec	850 hp
B	Worthington Pump	960	2300
C	Worthington Pump	1410	2550
D	De Laval	860	4100
E	Ingersoll-Rand	1210	6200


relative sound pressure level
db
overall
level
0
-10
-20
-30
-40
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps
compressor
compressor E
compressor A
compressor D

with rotation speeds from 1780 to 11,700 rpm. These machines differ considerably in the number of stages and blades. In general, different stages in a given machine have a different number of blades.

In a general way, the characteristic feature of compressor noise is a peaked spectrum with a maximum in a frequency range related to the rotation speed and the number of blades. In a sense, a compressor is a siren, but a rather complicated one. Different stages having different numbers of blades will generate different fundamental frequencies. The sound generated by any one stage will also have a rich harmonic content owing to non-linearities in the variation of air passage constrictions and owing to non-laminar flow conditions in the very high speed air streams around the blades. These effects combine to give a rather broad peak, with components extending over several octaves of frequency. The actual peak in the spectrum may be shifted somewhat from the calculated fundamental, owing to harmonic or sub-harmonic generation, or to variations of the compressor housing TL as a function of frequency.

The second major factor in compressor noise appears to be the total horsepower delivered to the compressors. On the basis of present available information, it appears that the total sound power increases about 6 decibels for each doubling of horsepower. This is surprising at first glance, since a doubling of acoustic power corresponds to a 3 decibel increase.

The steeper rise of 6 decibels per horsepower doubling means that axial-flow compressors, at least within the range studied, become increasingly efficient noise generators as their power increases. This seems plausible, at least up to some limit, on the basis of non-linear phenomena, such as shock waves or incipient shock wave behavior, at very high air speeds. However, this steep rate of increase seems to be associated

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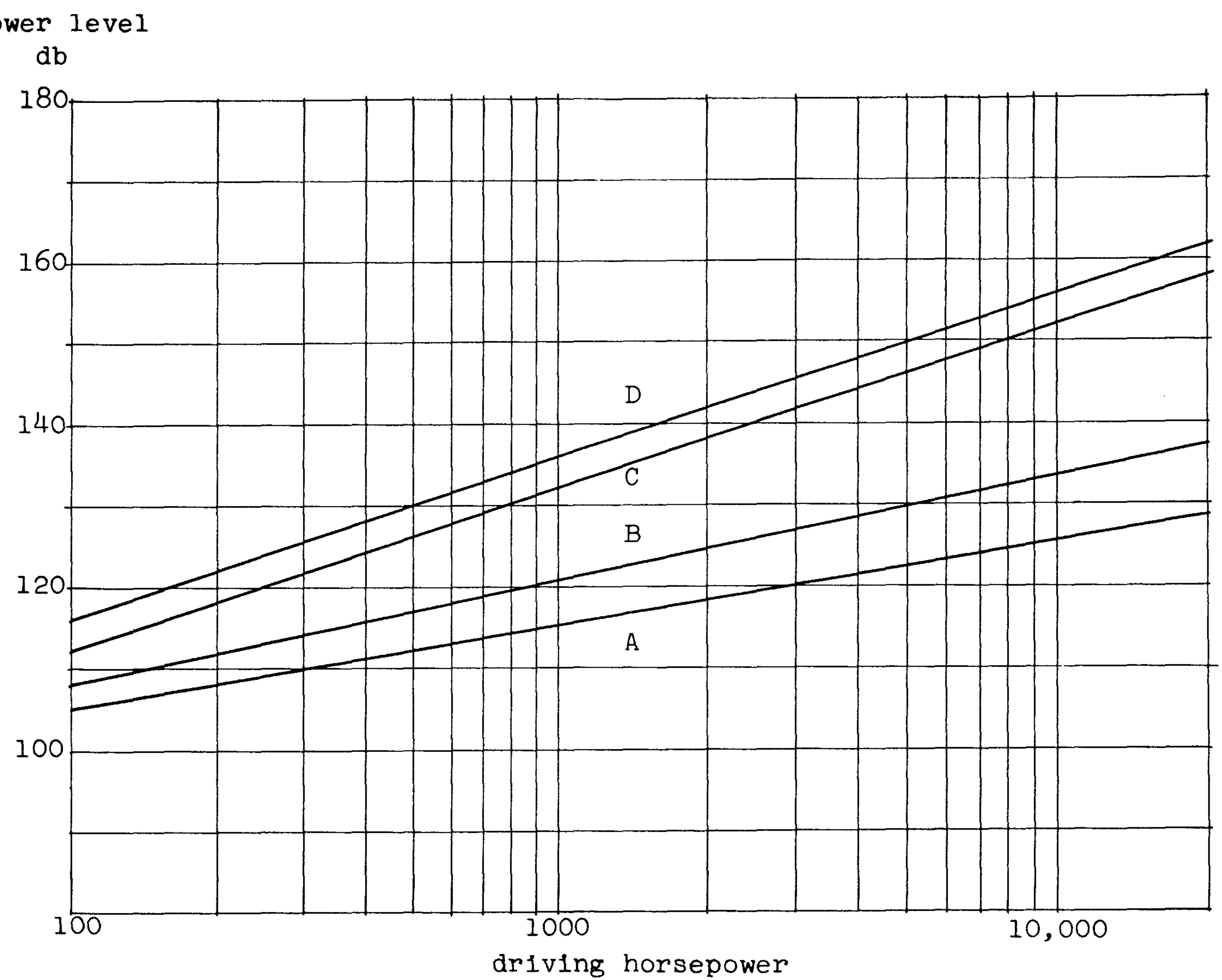
Figure 6.6

Measured noise spectra for the compressors indicated in Fig. 6.5. The ordinate gives the number of decibels in each frequency band relative to the overall SPL. For example, if the overall SPL at a certain point is 95 db for compressor D, then the power in the 150-300 cps band is 80 db.


power level
db
180
160
140
120
100
D
C
B
A
100
1000
10,000
driving horsepower

only with the noise in the peak frequency region which, as discussed above, derives from the siren-like action. At lower frequencies, in the neighborhood of 500 cps, the sound power increases about 4 db per doubling. The noise output at these lower frequencies is apparently associated with the more conventional mechanically generated vibrations.

The relations discussed above are given quantitatively in Fig. 6.7, which is a design chart derived empirically from available measurements on axial-flow compressors. The procedure for using this chart is as follows:

1. Determine the following characteristic frequencies

   F_M = frequency of maximum sound output

   = 1/30 x (compressor rpm) x (av. no. of rotor blades)

   F_H = highest frequency of significant output

   $= F_M^{\,2}/400$

   and locate these frequencies by interpolation on the octave band frequency scale.

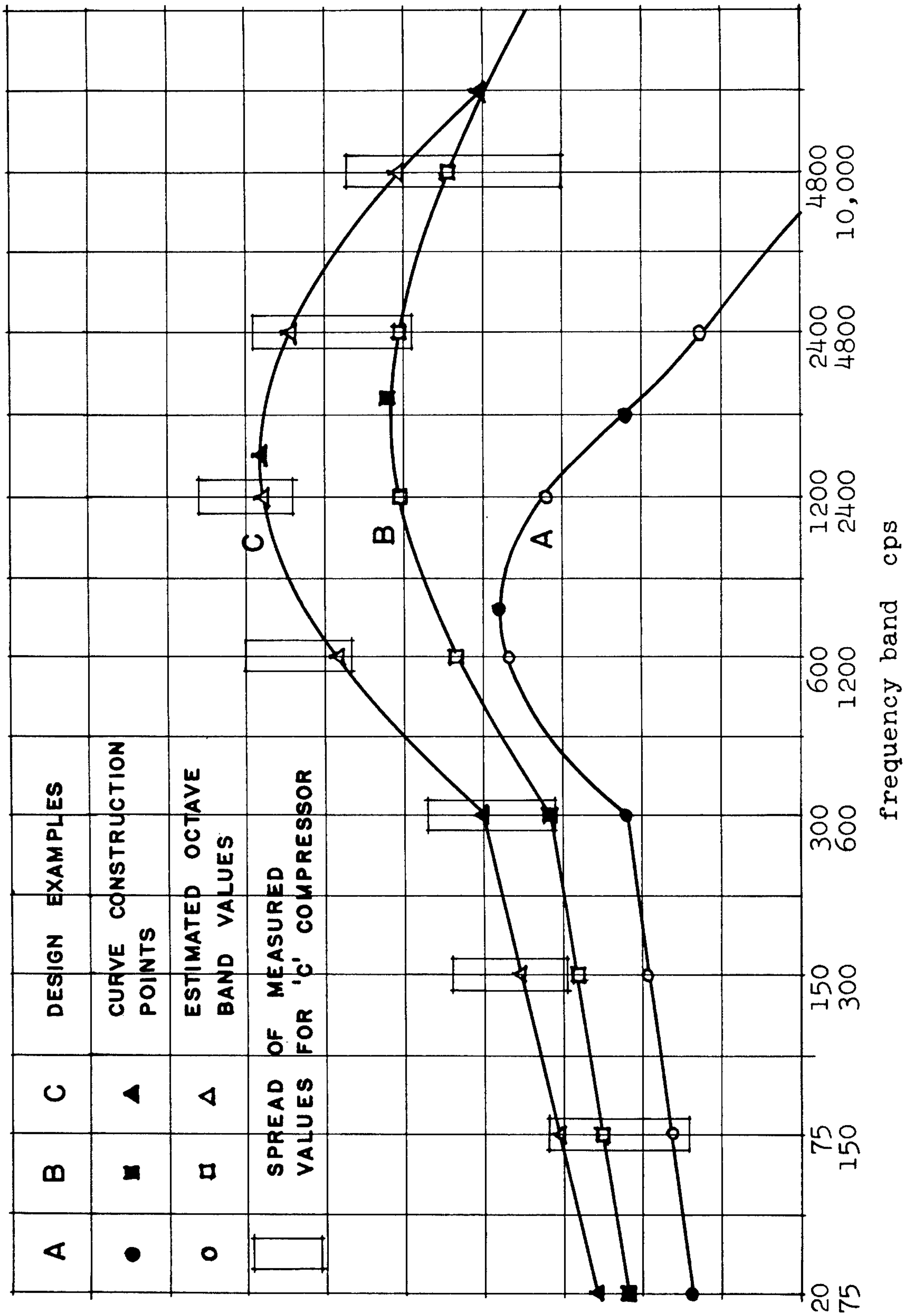
2. Using curve A in Fig. 6.7, enter the abscissa at the rated horsepower of the compressor and on the ordinate read the sound power level in the 20-75 cps band.

3. Similarly, using curve B, read the power level in the 300-600 cps band. This is also the power level at the above defined highest frequency F_H.

Figure 6.7

Empirical design chart for calculating overall power levels and spectra for the noise generated by axial-flow compressors when the compressor driving horsepower is known. The curves give

A. power level in 20-75 cps band
B. power level in 300-600 cps band
C. power level at frequency of noise maximum
D. overall power level.


A B C
DESIGN EXAMPLES
CURVE CONSTRUCTION POINTS
ESTIMATED OCTAVE BAND VALUES
SPREAD OF MEASURED VALUES FOR 'C' COMPRESSOR
A
B
C
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

4. Similarly, using curve C, read the power level at the above determined frequency F_M of maximum output.

5. Plot the above determined levels. Join the 20-75 cps point with the 300-600 cps point by a straight line. Draw a symmetrical curve through the points at 300-600, F_M and F_H. From this complete curve read the octave band values of power level.

6. The overall sound power level is read from curve D of Fig. 6.7, and is seen to lie 4 db above the level at F_M.

The above design procedure is illustrated by a set of curves given in Fig. 6.8. Calculations have been made for three axial-flow compressors.

Example	hp	F_M
A	300	1000
B	600	2500
C	2500	2000

In Fig. 6.8, curve construction points at the above discussed four frequencies are shown by solid symbols; the interpolated values are shown by open symbols.

Examples A, B, and C correspond to three cases that have been measured. The measured PWL for compressor C are shown in Fig. 6.8 by vertical datum bars that indicate the spread of the values obtained. The sound levels are generally different at different positions around a compressor; these variations are about $\pm$ 5 decibels in example C. Similar agreement was obtained for cases A and B. This empirical calculation method has also given values in rough agreement with measurements on an 11,200 hp centrifugal compressor.

Figure 6.8

Three examples illustrating the use of the curves given in Fig. 6.7 to calculate values of PWL. Measurements which were made on compressor C are indicated to show the expected difference between measured and calculated levels.

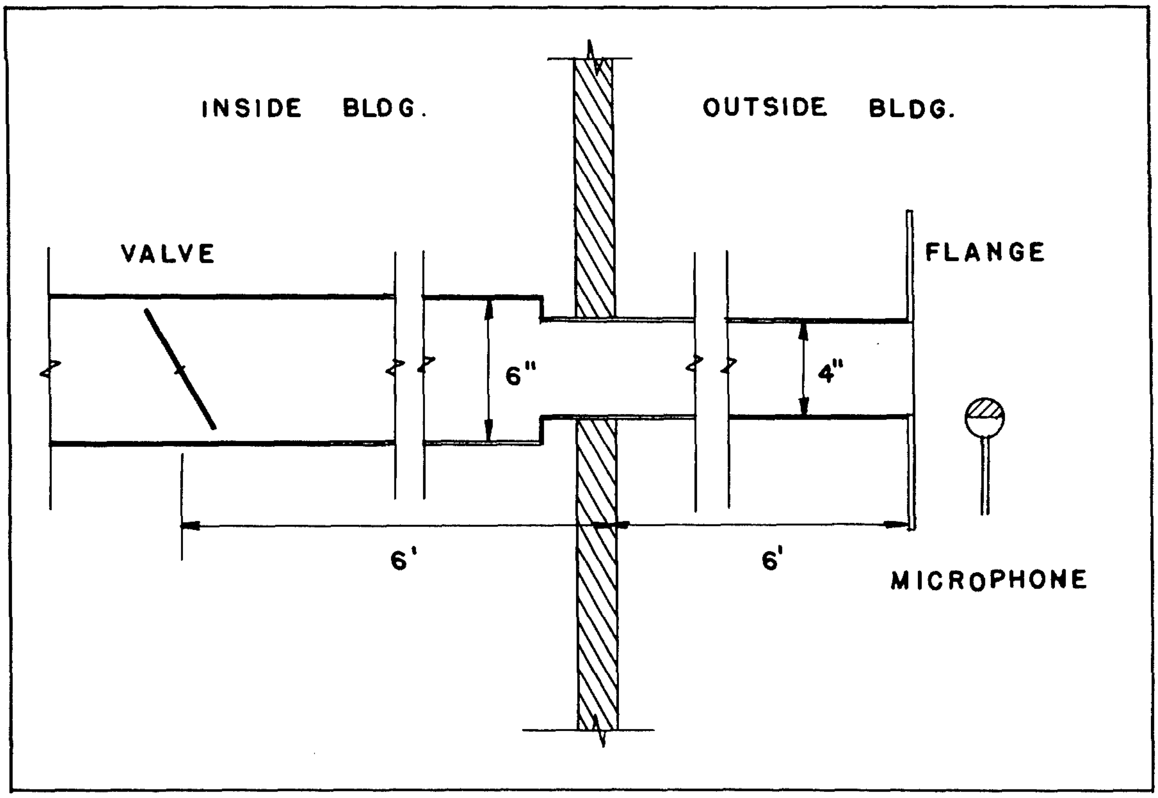


Figure 6.9

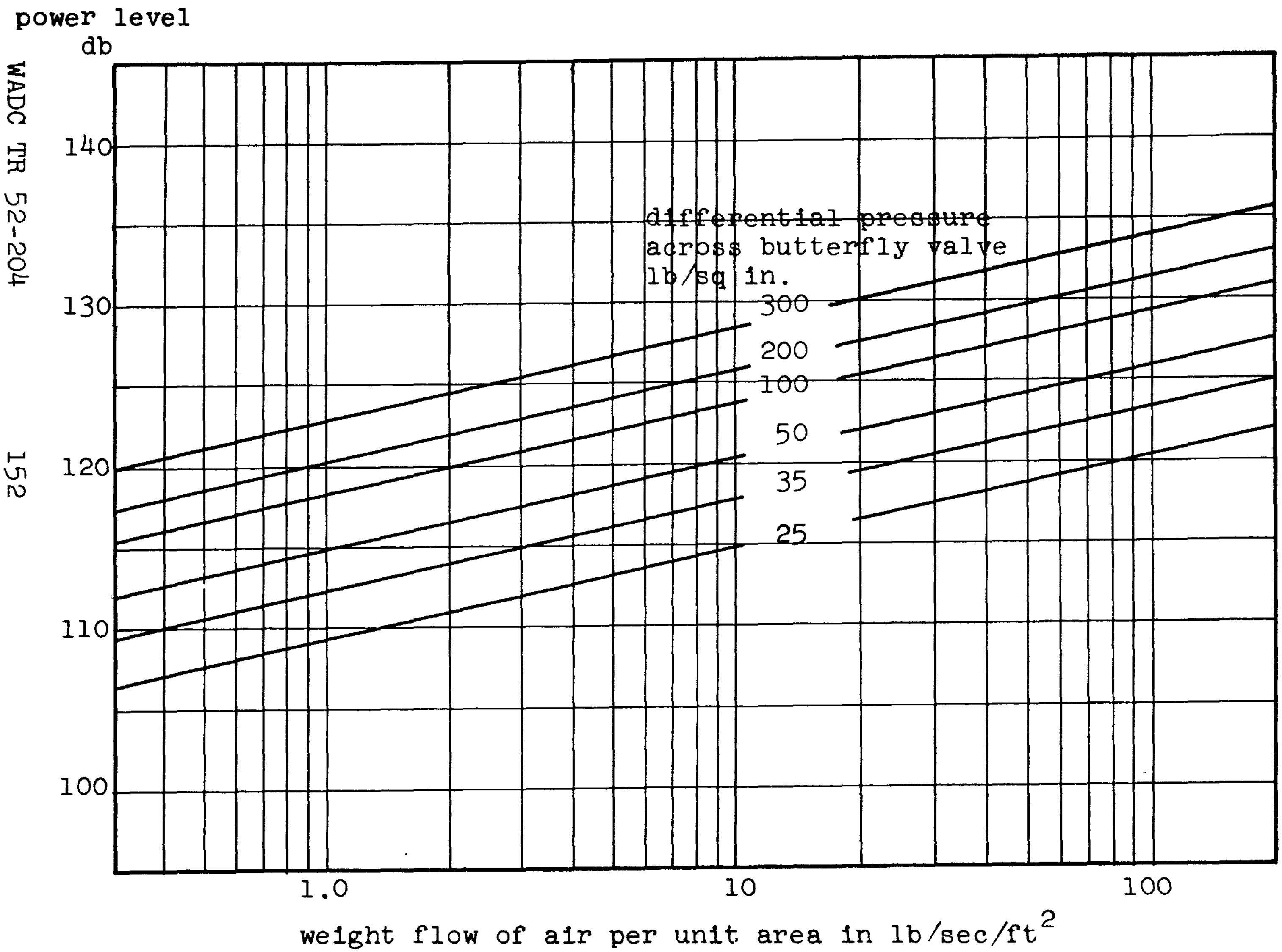
Butterfly-valve structure on which noise measurements were made.

6.4 Valve Noise

Noise measurements have been made on a circular disc-type butterfly valve over a wide range of mass flow of air and pressure differential across the valve. The sources of valve noise can best be discussed by considering the specific structures on which these measurements were made. Two main types of noise exist:

1. Air Noise
   - (a) Turbulence caused by the valve.
   - (b) The characteristic tone(s) caused by the passage of air through the small opening around the valve.
   - (c) Turbulence associated with the passage of air through the small opening.

2. Pipe Noise
   - (a) Pipe vibration caused by turbulence of air inside.


power level
db
140
130
120
110
100
differential pressure
across butterfly valve
lb/sq in.
300
200
100
50
35
25
1.0
10
100
weight flow of air per unit area in lb/sec/ft2

In an actual installation there may be pipe vibration caused by associated machinery. For the measurements taken here the valve was connected to a high pressure reservoir, thus eliminating this effect.

Figure 6.10 gives a series of curves of power level as a function of weight flow of air per unit area with pressure differential across the valve as a parameter. The plotted values of power level give one-half the total noise generated by the valve, i.e., the amount of noise radiated into the pipe in only one direction from the valve. The sound pressure level at the valve inside any given pipe of cross-sectional area A (in sq ft) can then be obtained by subtracting 10 log A from the power level. The sound pressure level thus obtained represents the overall SPL. The spectrum of the valve noise can be obtained from the curves given in Fig. 6.11, where a series of spectra are shown for different ranges of mass flow of air. For values in the ranges 0.1 to 0.5 and 0.5 to 5 lb/sec/ft^2, interpolation between the given curves is permissible.

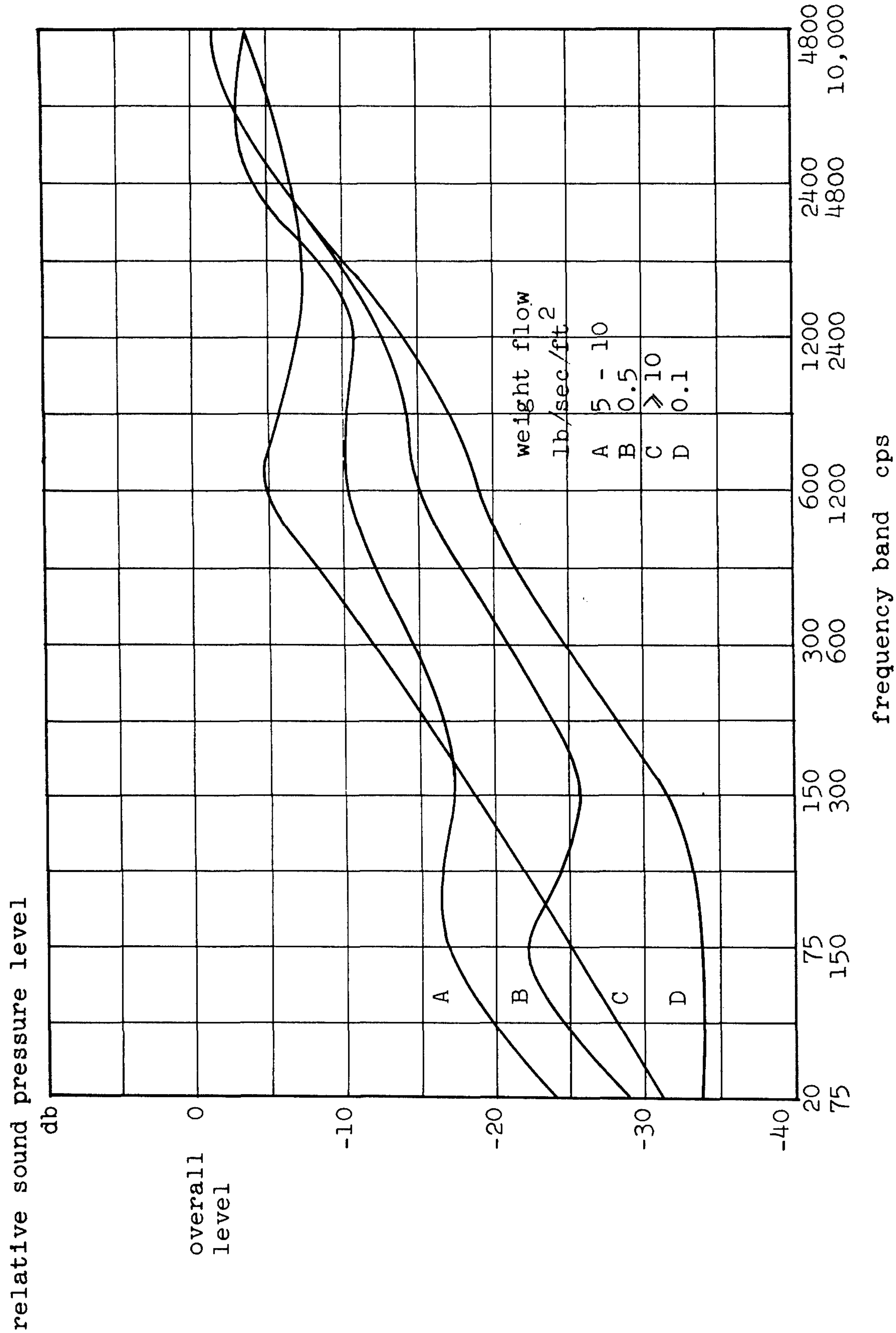
In establishing the curves of Figs. 6.10 and 6.11, no effort was made to isolate the various components contributing to the overall level. From the curves given in Fig. 6.11, however, note that the spectrum for values of air weight-flow greater than 10 lb/sec/ft^2 is typical of spectra obtained for high velocity air streams. This would indicate that above a certain weight-flow of air the noise is essentially air noise. Below a weight flow of air of 10 lb/sec/ft^2 the noise spectra in Fig. 6.11 show varying characteristics in the lower frequency ranges. These variations are attributed to the pipe noise component superimposed on the wind noise.

Figure 6.10

Power level of valve noise radiated in each direction as a function of the weight flow of air/unit area and the pressure drop across the valve. Since sound is radiated in both directions, add 3 db to obtain the total power level. Sound pressure level inside the pipe can be obtained from the relation

$$SPL = PWL - 10 \log A$$

where A is the cross-sectional area of the pipe, in sq ft.


relative sound pressure level
db
overall
level
0
-10
-20
-30
-40
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps
weight flow
lb/sec/ft 2
A 5 - 10
B 0.5
C ≥10
D 0.1
A
B
C
D

There is one condition of operation which is characterized by an unusually loud, high pitched whine ("screech" condition). This condition usually occurs for the case of small valve openings and results in a power level 10-15 db higher than would be estimated from Fig. 6.10.

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(4

(4) Simmons, L. F. G. and Salter, C., "Experimental Investigation and Analysis of the Velocity Variations in Turbulent Flow" Proc. Roy. Soc. A 145 212 (1934).

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(9) Goldstein, S., Modern Developments in Fluid Dynamics Vol. I p. 226.

Figure 6.11

Butterfly-valve noise spectra with weight flow as a parameter. The curves give the number of db below the overall SPL (or PWL) in each octave band.

CHAPTER 7

INDUSTRIAL MACHINE NOISE

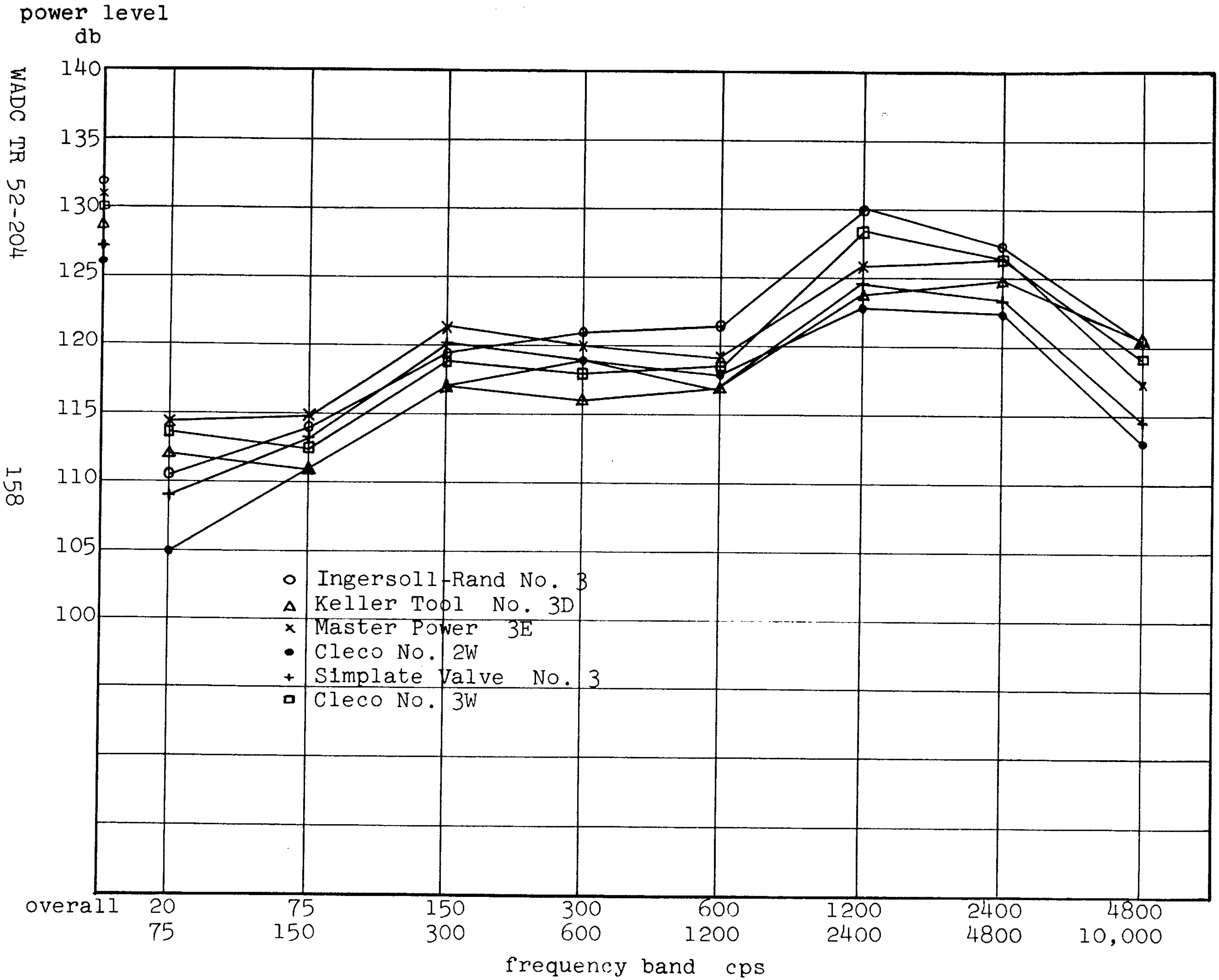
7.1 Introduction

This chapter presents data for a selected group of noise sources found in heavy industry. The chapter is subdivided into sections dealing with (1) noise from metal cleaning equipment and power tools, (2) noise from motors and motor-generator sets and (3) noises originating in drop forge plants. The acoustic information in the first two sections is given in terms of power level spectra while in the third section, since the radiating area is not determined, the information is given in the form of idealized graphs of overall sound pressure levels as a function of time. These graphs were drawn from oscillograms taken of the transient noise levels of drop forge hammers.

7.2 Noise from Metal Cleaning Equipment and Power Tools

This section contains some generalized information 1/ on noise from conventional commercial cleaning room equipment: chipping hammers, grinders, arc welders, oxyacetylene torches, and small pneumatic wrenches. The information is given in terms of power level spectra in Figs. 7.1 to 7.4. From these curves one can obtain the sound pressure level at a specified distance from the tool, in a room with a specified amount of sound absorption, by using the curves presented in Fig. 3.3. This figure 2/ is made on the basis of uniform spherical radiation from the tool, a condition that is reasonably well satisfied for all tools reported here, at distances of 2 ft or more from the tool.

To illustrate the transformation of the information, consider a specific example. Let us find the highest expected sound level in the 1200-2400 cps band, at a distance of 5.5 ft from a pneumatic chipping hammer operating in a room with an area and absorption corresponding to a room constant R = 1000. From Fig. 3.3 we find for a distance r = 5.5 ft, and room constant R = 1000, a correction of -22 db to convert from power level to sound pressure level. In Fig. 7.2 we find from the top curve, a power level of about 130 db in the 1200-2400 cps band. Therefore, SPL = 130 - 22 = 108 db in this band.



power level
db
140
135
130
125
120
115
110
105
100
overall
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps
o Ingersoll-Rand No. 3
Δ Keller Tool No. 3D
x Master Power 3E
• Cleco No. 2W
+ Simplate Valve No. 3
□ Cleco No. 3W

Similarly, we could compute the levels in all bands, and then combine them (by energy addition) to obtain an overall sound level. In the above example we would compute an overall SPL of about 110 db for the highest level expected from chipping hammers at r = 5.5 ft in a room with R = 1000. This value agrees with the upper limit of values given for chipping hammers in Table 7.1. This table, for convenience, contains the overall values of (1) power level in db re 1.0×10^{-13} watts; (2) power in watts, and (3) sound pressure level in db re 0.0002 dynes/cm^2 at 5.5 ft from the tool in a room with R = 1000 units.

Many other cases of practical interest can be examined by further application of data along the lines illustrated above. The remainder of this section discusses special features of the noise from different types of cleaning tools.

Types of Tools. The information compiled in this section is based on measurements on several types and varieties of tools as follows:

1. Chipping hammers (six types)
2. Swing frame grinders (one unit with 16 in. and 18 in. grinding wheels
3. Air grinders (four types and four different grinding wheels)
4. Arc welders (two types)
5. Oxyacetylene torches (three types)
6. Electric grinders (two types and three different wheels)
7. Pneumatic wrenches (one type).

Each class of equipment was investigated under a variety of operating conditions which represent the total range normally encountered in practice. All of the tools were operated by an experienced operator performing typical tasks, but under supervised conditions suitable for the purposes of the investigation.

The overall noise information for each type of tool is summarized in Table 7.1.

Chipping Hammers. The results of measurements on chipping hammers are plotted in Figs. 7.1 and 7.2. Fig 7.1 shows the

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Figure 7.1

Power level spectra for chipping hammers of various makes, operating at 90 psi gauge on a steel plate 18 x 18 x 3/4 in.

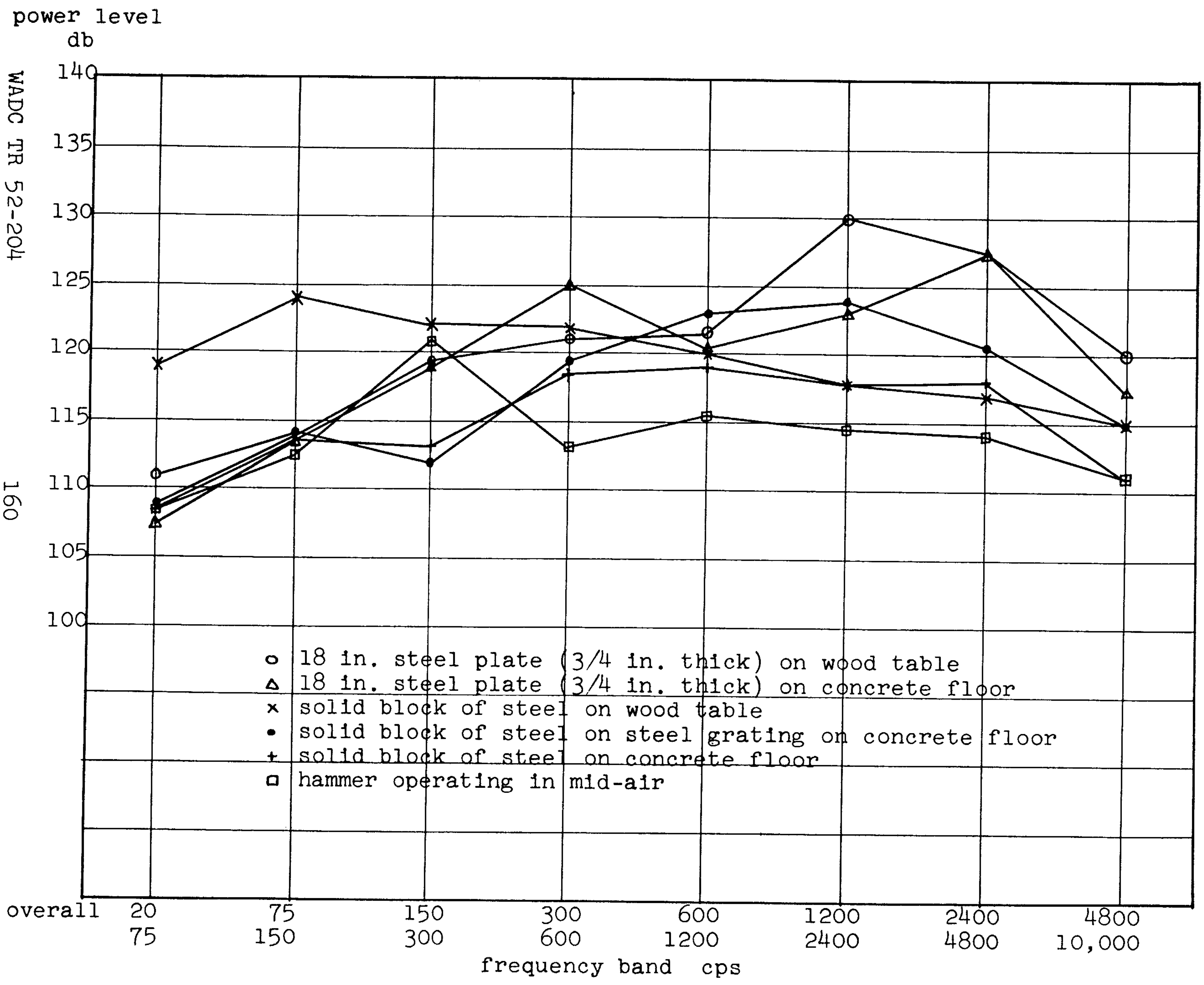

power level
db
140
135
130
125
120
115
110
105
100
o 18 in. steel plate (3/4 in. thick) on wood table
Δ 18 in. steel plate (3/4 in. thick) on concrete floor
× solid block of steel on wood table
• solid block of steel on steel grating on concrete floor
+ solid block of steel on concrete floor
□ hammer operating in mid-air
overall
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

TABLE 7.1

NOISE OUTPUT FROM CLEANING ROOM EQUIPMENT

Tool	Power Level in db re 1.0×10^{-13} watts	Acoustic Power in Watts	Overall Sound Level in db at 5.5 ft in room with 1000 units of absorption
Chipping hammers Normal Operation at 90 psi gauge	126-132	0.3-1.5	104-110
Portable air grinders- Steady operation at 90 psi gauge	109-115	0.006-0.03	87-93
Swing Frame Cut-off Grinders - Normal Operation 16 in. and 18 in. Wheels	116-122.5	0.03-0.15	94-100.5
Electric Grinders - Normal Operation 7 in. and 9 in. Wheels	111-116	0.01-0.04	89-94
Oxyacetylene Torches	102.5-111	0.0015-0.015	80.5-89
Pneumatic Wrench (small size)	116-122	0.03-0.15	94-100

power spectra in octave bands for the six hammers working on a steel plate (18 in. x 18 in. x 3/4 in.). The overall value for the combined spectra is given to the left. It will be noted that there is little difference in the overall PWL of the different makes of hammers.

Figure 7.2

Power level spectra for Ingersoll-Rand #3 chipping hammer operated at 90 psi gauge under various circumstances.

Figure 7.2 shows the change in spectra for one hammer (Ingersoll-Rand #3) working on different materials. The highest overall level occurs for the hammer operating on a steel plate located on top of a wooden table. The same hammer operating on a partially hollow steel block in the same position produces a spectrum with reduced high frequency energy content. When the block is placed on a concrete floor, the overall level and high frequency components are even further reduced.

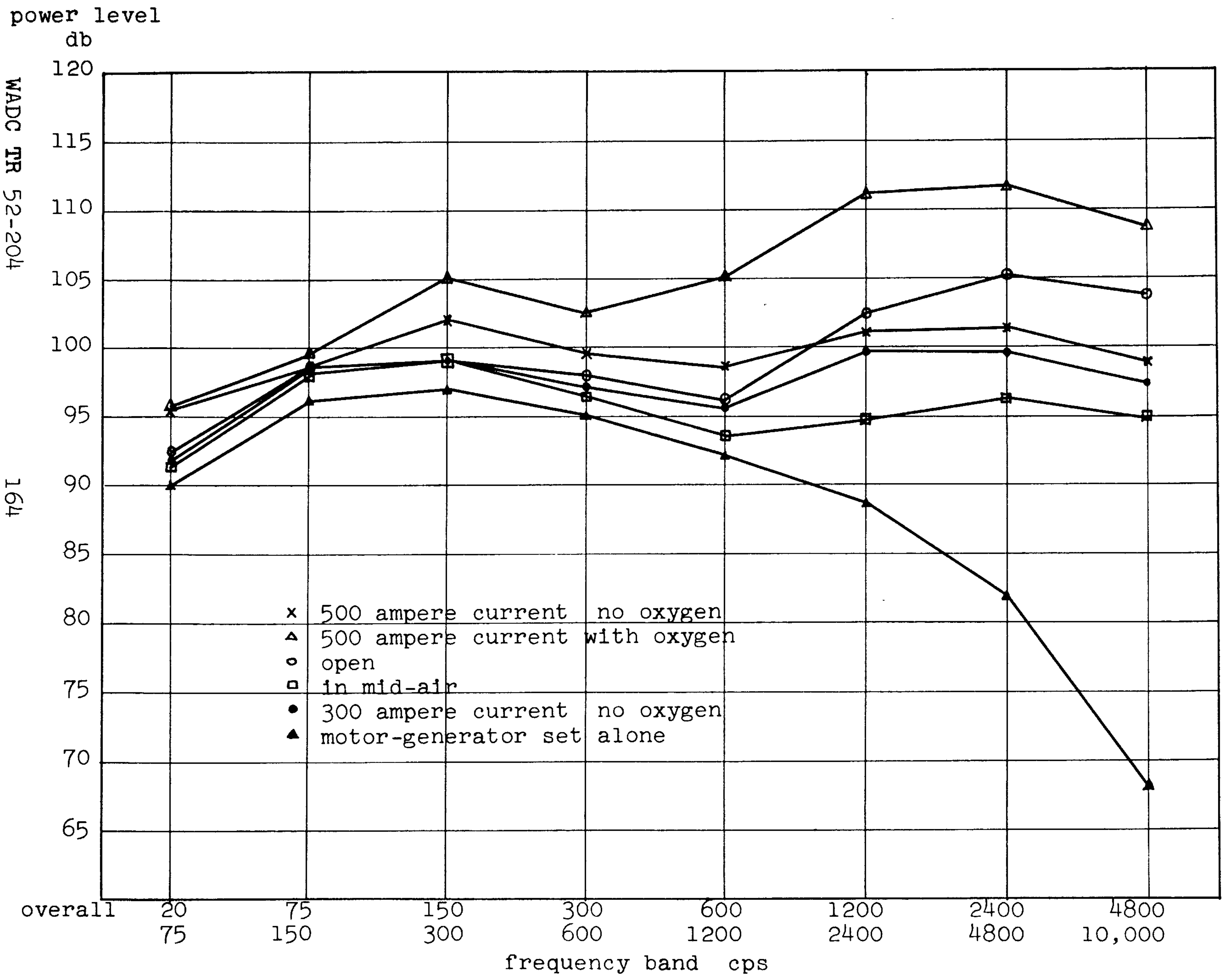
Portable Air Grinders. The power level of portable air grinders operating on steel plate is about 112 db, considerably below that of the chipping hammers. There is little difference in the overall PWL regardless of whether the equipment operates at 4500 or 9000 rpm or whether a cup wheel or disc wheel is used. Operating pressure applied to the various air grinders changes the values in the frequency bands but has very little effect on the overall values.

Portable Electric Grinders. Measurements have been made on two portable electrically-operated grinders which were adapted for 7 and 9 in. grinding wheels. Studies were made for both conditions of steady operation (with the operator exerting a more or less constant pressure on the work) and normal operation (with the operator moving the tool back and forth over the surface of the work in the usual manner).

The portable electric grinders show approximately the same power level as the air grinders, from 111 to 116 db, depending on the type of wheel and differences in equipment. There is little difference in the overall noise level regardless of the condition of the plate being ground.

Oxyacetylene Torches. Three different types of oxyacetylene torches have been studied. An Oxweld C-32 blow pipe was used with three different nozzles (Nos. 4, 8, and 12). The size of the nozzle in turn determined the amounts of oxygen and acetylene consumed by the torch.

The results show that the preheat or oxygen-starved condition, which is used to raise the metal to a temperature near its melting point, is the quietest operating condition. Once the oxygen is applied to the torch, the energy content of the higher octave bands increases and this, in turn, causes the overall level to increase by 6 to 15 db. In none of the operating conditions does the overall power level exceed 111 db.



power level
db
120
115
110
105
100
95
90
85
80
75
70
65
overall
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps
× 500 ampere current no oxygen
△ 500 ampere current with oxygen
○ open
□ in mid-air
● 300 ampere current no oxygen
▲ motor-generator set alone

The cutting operation varied from 111 db power level for the large No. 12 nozzle to 103 for No. 8 nozzle and 97 for No. 4 nozzle. In other words, the larger the nozzle size the greater the noise level.

Electric Welders. Measurements have been taken on electrodes for defect removal, operated with and without an oxygen jet. The metal electrode without the oxygen jet produces an overall power level of 104 to 107 db regardless of the variation in electrode size from 1/8 to 1/4 in. diameter or in the variation of the operating current from 300 to 500 amperes.

The use of the carbon electrode and oxygen results in the spectrum shown in Fig. 7.3. The application of oxygen substantially increases the levels in the octave bands above 150 cycles per second, and with 500 amperes in the arc, the overall level is increased by 8 db. In all cases the overall level of the noise generated by the motor-generator set, which supplies the electrical energy for the welder, is only a few decibels below that of the arc itself.

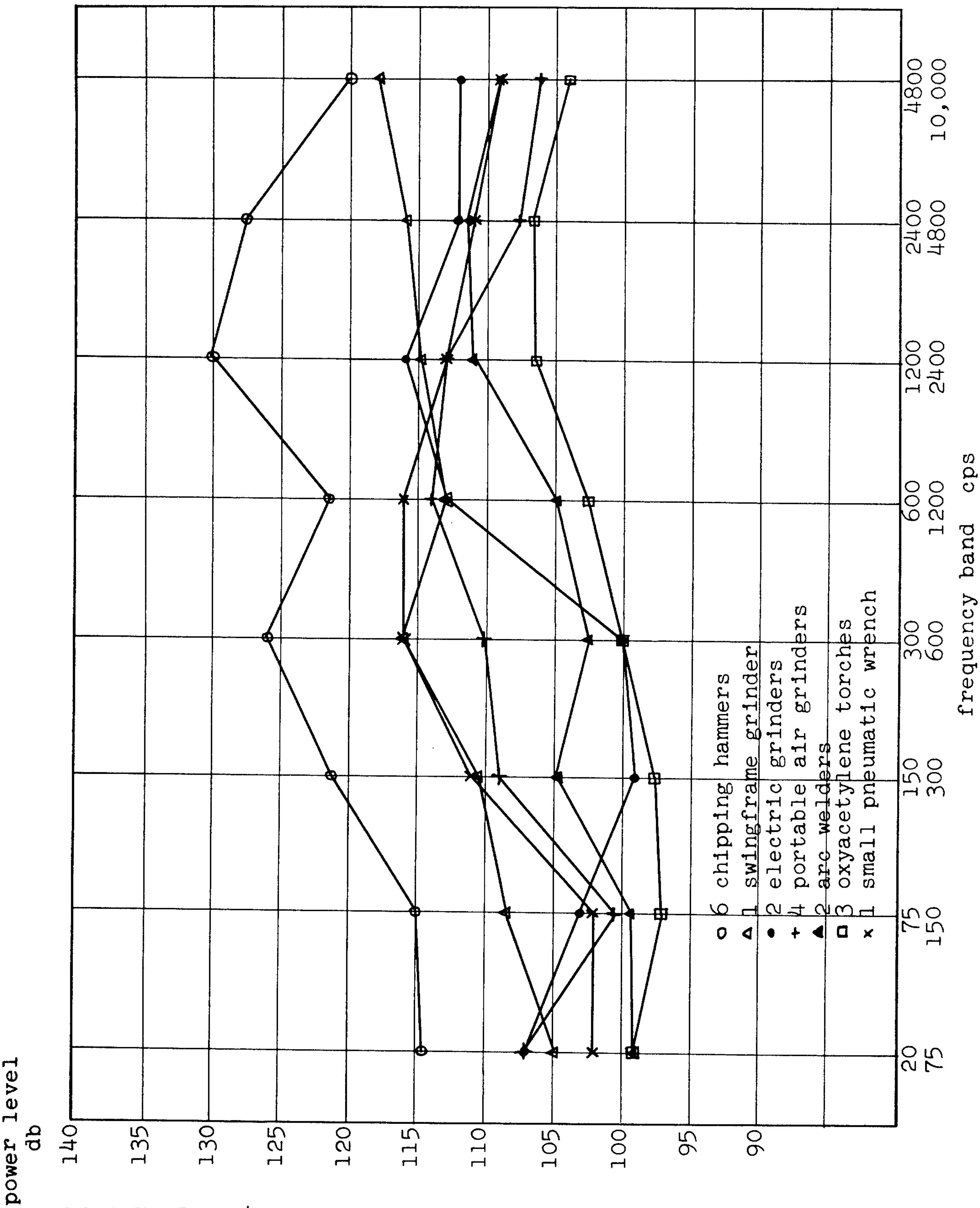
The octave band levels of the motor-generator set decrease with frequency whereas those of the carbon arcs increase in the higher bands. The result is that the higher frequency components generated by the arc are clearly audible over the predominantly low frequency noise of the motor-generator set. A sputtering noise occurs under certain conditions when the arc is formed and molten metal is thrown out from the working area. This sputtering causes the levels in each band to vary by 3 to 5 db.

Swing Frame Grinders. Measurements have been made on a swing frame cut-off grinder manufactured by Fox Grinders, Inc., having a Westinghouse motor of 10 horsepower and 1750 rpm. This unit runs at a single speed. Measurements were taken on a 16 in. wheel and an 18 in. wheel.

Pneumatic Wrench. Measurements were made on a small pneumatic wrench at a distance of 3 ft in a large room while the wrench was being used to set bolts on the rim of a large steel tank.

Figure 7.3

Power level spectra for a G.E. Electric Arc Welder Type WD 34 with the Kwik-Arc Jet Torch (carbon arc and oxygen jet) operating under various conditions.



power level
db
140
135
130
125
120
115
110
105
100
95
90
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps
o 6 chipping hammers
Δ 1 swingframe grinder
• 2 electric grinders
+ 4 portable air grinders
▲ 2 arc welders
□ 3 oxyacetylene torches
× 1 small pneumatic wrench

Comparison Curves. The power levels of all cleaning room equipment are shown together in one illustration, Fig. 7.4. In this figure there has been plotted, for each class of tool involved, the highest level observed in each frequency band under the noisiest operating conditions. Thus, the curves in Fig. 7.4 do not represent any single tool but rather the highest level expected on each class of tool. It is believed these curves give the most significant generalization, since they indicate the most severe condition likely to be encountered with any tool of a given class used under any condition.

7.3 Noise from Induction Motors and Motor-Generator Sets

Generalized information is given in this section on noise from induction motors, of the type equipped with heavy-duty casings for use outdoors, and from large motor-generator units mounted inside a building. The information is given in terms of sound pressure level spectra in Figs. 7.5 to 7.7. The power level for these units cannot be calculated because of a lack of data concerning the radiating area of the source or its location in a specified room of known size. The sound pressure levels are given for specific distances and locations with respect to the particular source.

Induction Motors. The curves shown in Fig. 7.5 and 7.6 are both for induction motors in use in places outside of buildings where they are exposed to the weather. These motors were all equipped with heavy-duty casings. All measurements represented by the curves in these two figures (which differ mainly in the horsepower rating of the motors) were made with a microphone positioned two inches from the open end of the casing.

In Fig. 7.5 are shown the levels from the smaller motors with such variables as horsepower, speed and manufacturer being designated on each curve. It can be seen from an examination of the four curves representing data at 3600 rpm that the horsepower rating is not an indication of the noise levels to be expected. There is some slight indication that a variation in speed might cause a corresponding variation in noise levels but the data are insufficient to confirm this conclusion.

Figure 7.4

Power level spectra for various cleaning room equipment under the noisiest operating conditions.

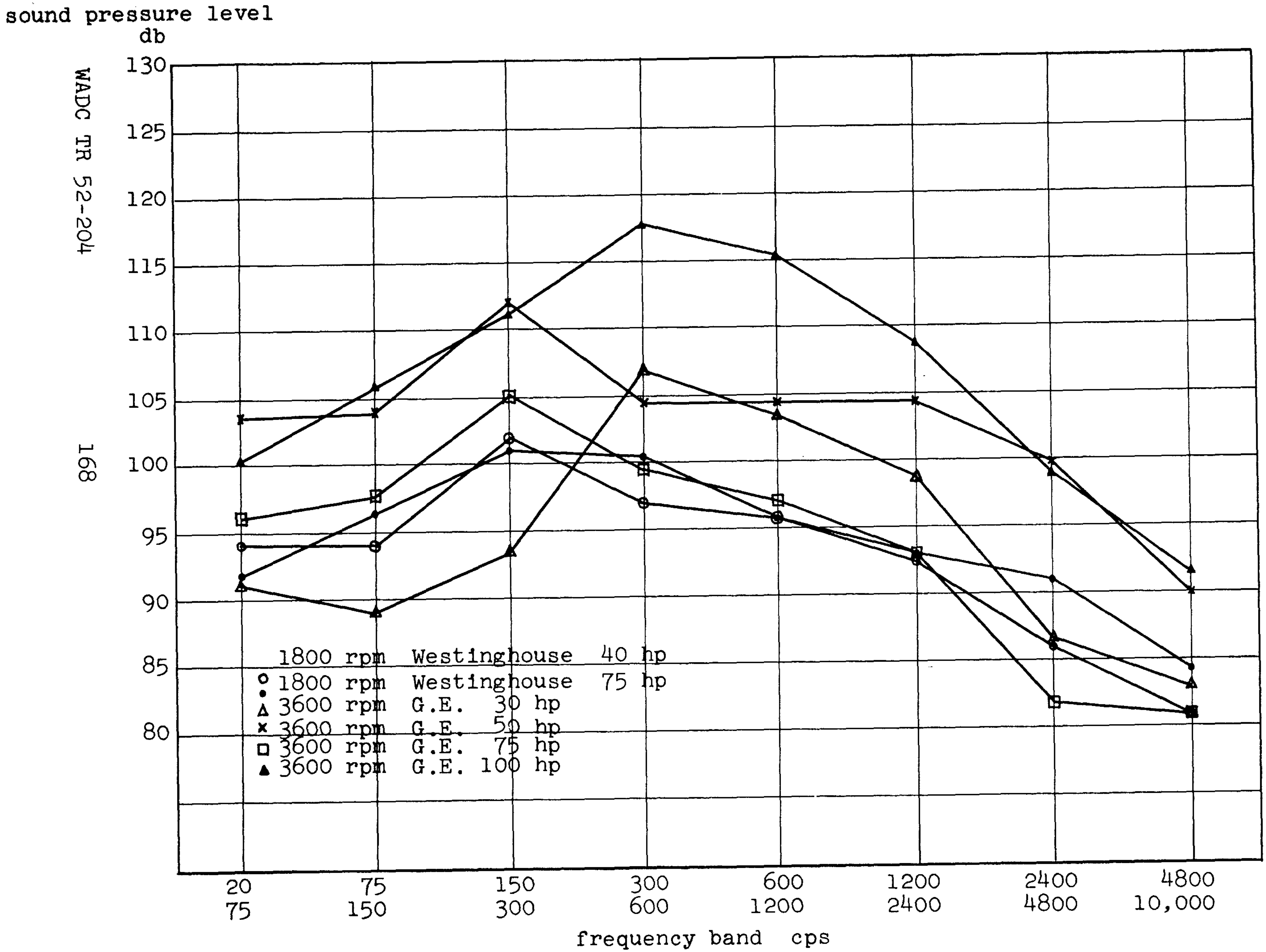

sound pressure level
db
130
125
120
115
110
105
100
95
90
85
80
1800 rpm Westinghouse 40 hp
1800 rpm Westinghouse 75 hp
3600 rpm G.E. 30 hp
3600 rpm G.E. 50 hp
3600 rpm G.E. 75 hp
3600 rpm G.E. 100 hp
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

Figure 7.6 illustrates the spectra measured 2 in. from the open end of the large induction motors of the same general type as illustrated in Fig. 7.5, and equipped with the same heavy-duty casings. Here again there appears to be no connection between the noise levels and the variable parameters of horsepower and manufacturer.

Motor-Generator Units. A few measurements are available for very large motor-generator units. Figure 7.7 shows the spectrum as measured on the centerline of a 1750 hp unit at a distance of 3 in. from the flywheel. Other pertinent data on this unit are that the generator speed was fluctuating from 846-900 rpm, and that the generator is a 12-phase alternator supplying current at 56.4 cycles. The motor unit requires 12,500 cfm of cooling air while the generator requires 40,000 cfm.

7.4 Noise in Drop Forge Shops

The main sources of noise in a drop forge shop are of two types. Noise from the furnaces and blowers is steady with time and thus established procedures may be used to measure the noise and evaluate it with respect to potential damaging effects. The drop hammers, however, radiate transient sounds for which measurement procedures as well as the criteria for risk of damage to the hearing mechanism (see Volume II) are less well established.

The measured characteristics of each of the two classes of noise are presented and discussed separately in this section.

Steady Noise in a Forge Shop from Furnaces and Blowers. The results of an octave band analysis of the sound in the vicinity of two typical furnaces are shown in Fig. 7.8. The curves show how the sound energy is distributed with frequency at locations 5 ft in front of the furnace openings. In the general shop area the sound pressure levels may be about 11 db below the values in Fig. 7.8 for rooms where the reverberation time is of the order of 3 to 5 seconds. However,

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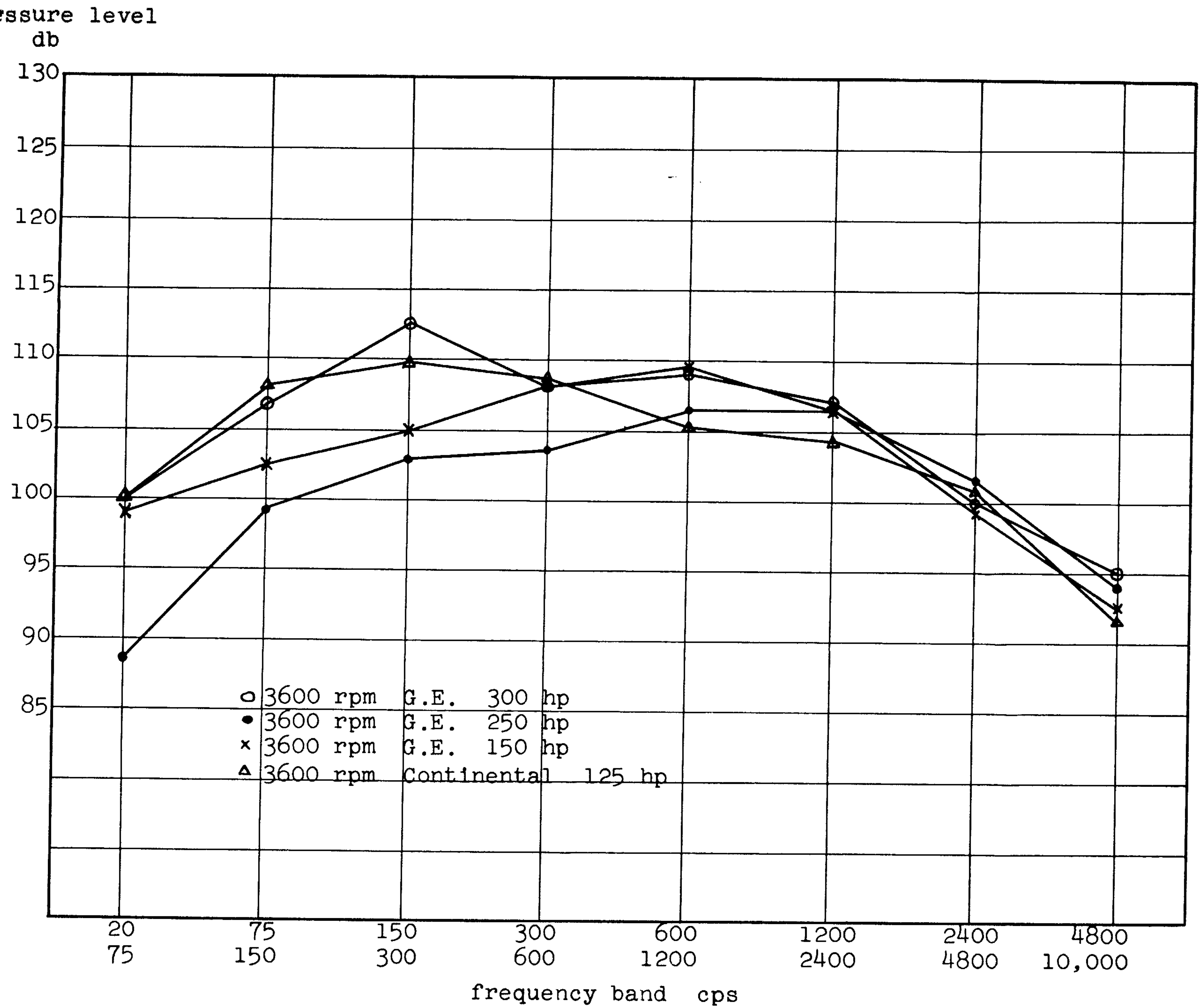
Figure 7.5

Noise spectra measured 2 in. from the open end of small, outdoor type induction motors with heavy-duty casings. Various speeds, horsepower and manufacturers are represented.


Pressure level
db
130
125
120
115
110
105
100
95
90
85
○ 3600 rpm G.E. 300 hp
● 3600 rpm G.E. 250 hp
× 3600 rpm G.E. 150 hp
△ 3600 rpm Continental 125 hp
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

with several furnaces operating (say 10), the average levels are increased by about 10 db over the corresponding values for one furnace only. Thus the levels in the shop area due to the furnaces above are only about 1 db below those measured at a distance of 5 ft from an average furnace.

Noise characteristics of the blowers associated with each furnace are described by octave band data in Fig. 7.9. The measurements were made in the same shop area as for the furnaces with 1 to 23 blowers operating simultaneously. The noise from 23 blowers represents the worst condition in this particular shop.

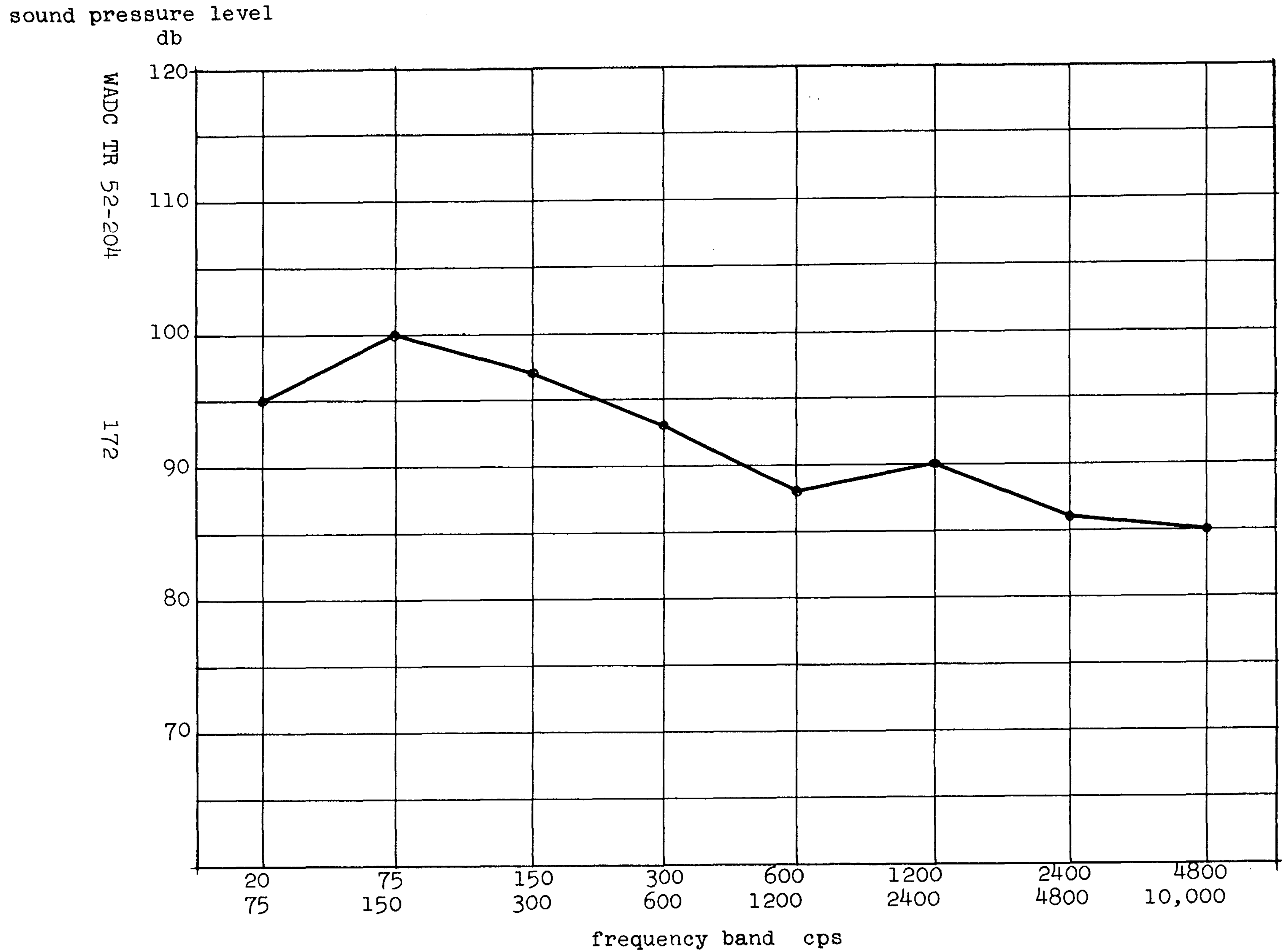
Impact Noise in a Forge Shop from Drop Hammers. Because they are transient in nature, the sounds from the drop hammers cannot be described adequately by the conventional procedure of octave band analysis in which the deflections of a meter are recorded. The measuring instruments must operate sufficiently rapidly in order to follow the changes in sound pressure caused by the hammer blow. An oscilloscope provides a suitably rapid indicator for the analysis of transient sounds.

Preliminary studies of oscillograms of the sound from a hammer blow indicate that the peak sound pressure levels are much greater than those from furnaces and blowers, and are well up in the range of damage-risk. An oscillogram of the variation of sound pressure at the operator's position for a typical blow from an 800 lb hammer was taken and an average decay envelope out to 800 milliseconds was traced from it; this is shown in Fig. 7.10

The following properties are indicated in the decay curve Fig. 7.10. Within 2 or milliseconds after the impact, the sound pressure rises to a peak value of about 1500 dynes/cm^2 for this type of hammer, and after another 1 or 2 milliseconds drops to about 1200 dynes/cm^2. During the next 20 milliseconds, the peak values of sound pressure decay at a rapid rate to about 280 dynes/cm^2 and subsequently the decay rate becomes slower.

Figure 7.6

Noise spectra measured 2 in. from the open end of large outdoor type induction motors with heavy-duty casings. Various horsepower and manufacturers are represented.


sound pressure level
db
120
110
100
90
80
70
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

The first 50 milliseconds of decay are attributable to the sound source alone, since no reflections have yet been received from the walls of the room. This initial decay is associated with the damped free vibrations of mechanical parts of the hammer, e.g., the die blocks. The small peak of sound pressure at time = 300 milliseconds represents the noise made by the hammer when it returns to its original position in preparation for a new drop.

The decay curve Fig. 7.10 has been smoothed and replotted as Fig. 7.11. In this figure, the rms sound pressure is plotted in decibels re 0.0002 dynes/cm^2. The rms sound pressure is taken at $1/\sqrt{2}$ times the peak value (3 db less than the peak value). The initial decay in sound pressure up to about 50 milliseconds is approximated by an exponential in Fig. 7.10 or by a straight line in Fig. 7.11. The rate of this initial decay is about 300 db/sec. After 100 milliseconds the decay rate changes to a new, slower value of about 20 db/sec, since the reverberation time of the room is about 3 seconds.

Figure 7.11 represents the idealized decay curve for a specific hammer. Oscillograms from several other hammers reveal the same general decay rates as those shown in this figure. The sound pressure level of the initial peak at the operator's ear position is in the range of 138 $\pm$ 5 db depending on the type and rating of the particular hammer. If the room characteristics were altered, the initial peak and decay rate for a given hammer would not change, since these are determined by the source alone, but the final decay rate would be different. The absolute level of the section of Fig. 7.11 with the 20 db per second decay rate is independent of the position in the room at which the sound is measured, providing sufficient time has elapsed to permit the establishment of a reasonably uniform distribution of sound energy throughout the room following the hammer blow.

Room Levels with Several Hammers Operating. Probably the most severe typical noise condition encountered by an operator of a drop hammer could be described as follows: his own hammer operates continuously at a rate of about one drop per second, at the same time other hammers at various locations in the

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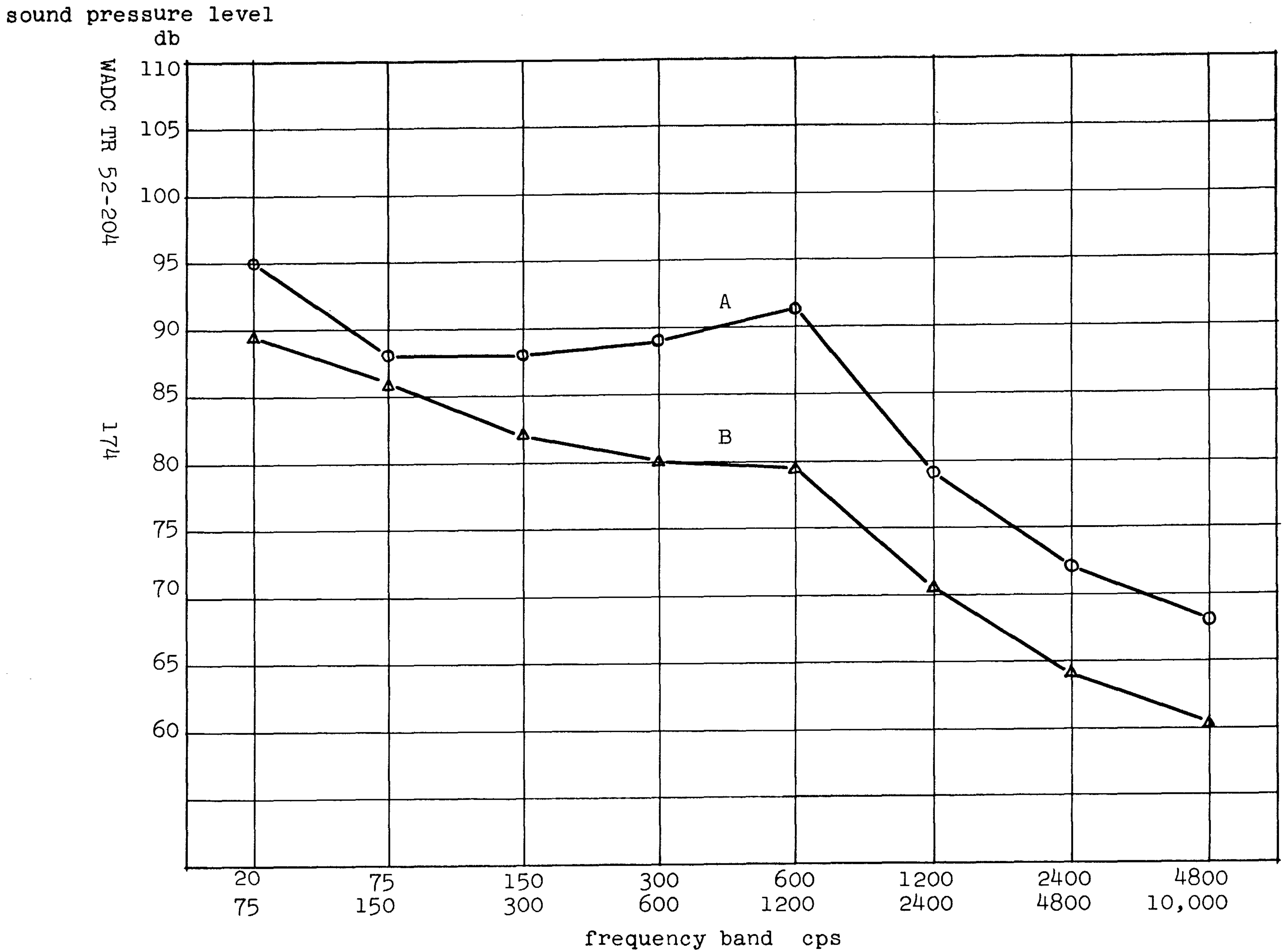
Figure 7.7.
Noise spectra taken near centerline of motor-generator set at a distance of 3 in. from the flywheel. The motor is a 1750 hp Westinghouse unit operating at 6000 volts, 7000 amp with 12,500 cfm cooling air. The generator is a Westinghouse 12-phase alternator varying between 846 and 900 rpm and supplying current at 56.4 cycles. 40,000 cfm cooling air required (10 blades on compressor rotor).


sound pressure level
db
110
105
100
95
90
85
80
75
70
65
60
A
B
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

shop make a total of, say, three impacts per second on the average. Assuming a uniform temporal spacing of four blows per second, the overall sound pressure levels existing at the operator's ear as a function of time would be shown as in Fig. 7.12. The strongest peaks (138 db for an average hammer) arise from the operator's hammer and peaks ranging from 110 to 121 db are received from other hammers, depending on the location and size of the hammer. In this idealized situation, there is no period longer than 250 milliseconds during which there is no hammer blow. Consequently, the sound pressure level in the room never drops below about 103 db as shown.

In the general shop area (not in the operator's position) a similar variation of sound pressure with time will be observed, assuming the same condition of four hammer blows per second. However, the peaks are not as high since the distance to the nearest forge may now be greater than 10 ft. The application of acoustical treatment to the room will not serve to reduce the peak values but will lower SPL between peaks as much as 20 db.

Figure 7.8

Noise spectra measured 5 ft from openings of
(A) back fired forge shop furnace
(B) side fired forge shop furnace

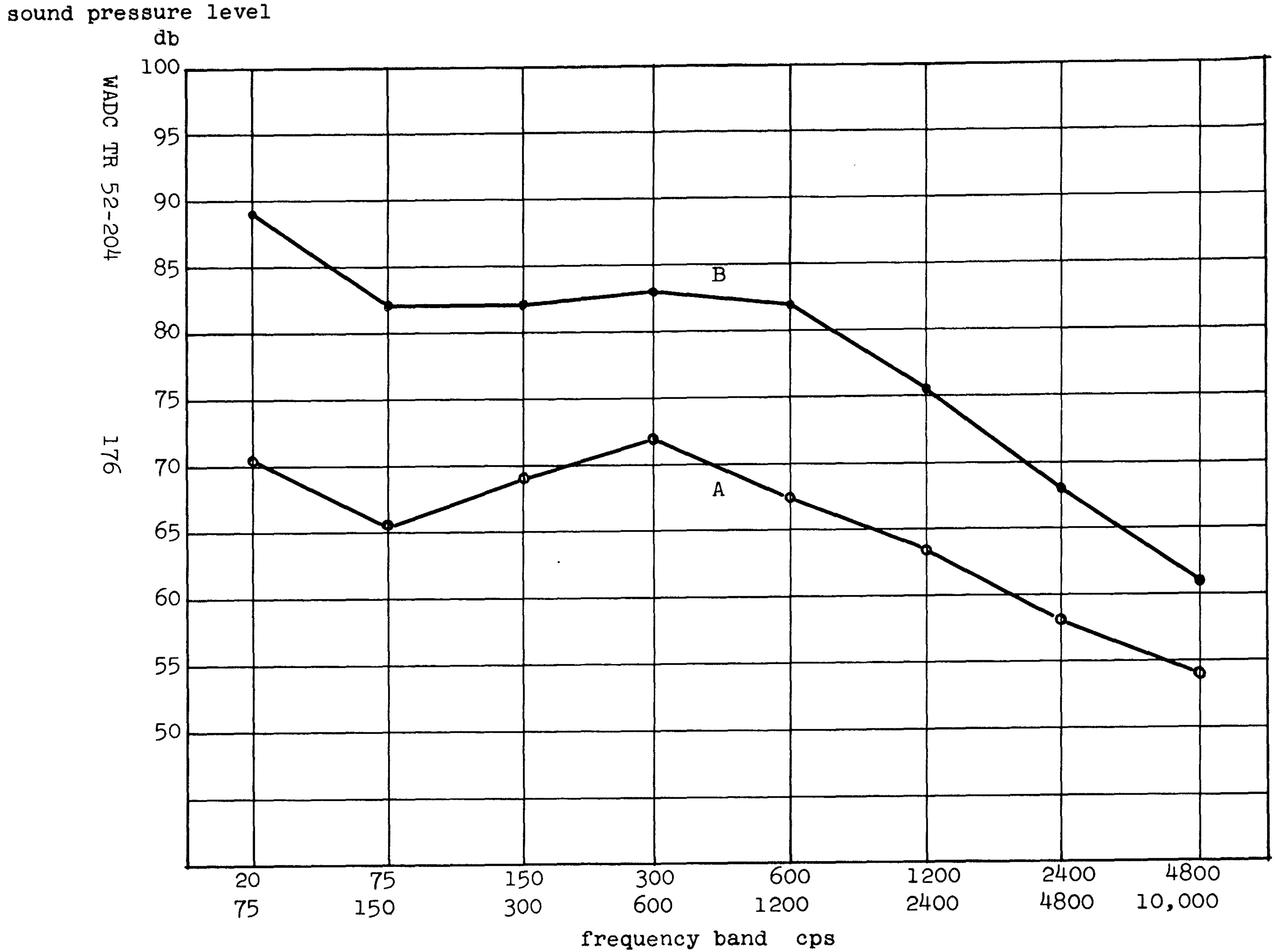

sound pressure level
db
100
95
90
85
80
75
70
65
60
55
50
B
A
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

Figure 7.9

Noise spectra from forge shop furnace blowers

(A) one blower operating

(B) twenty-three blowers operating

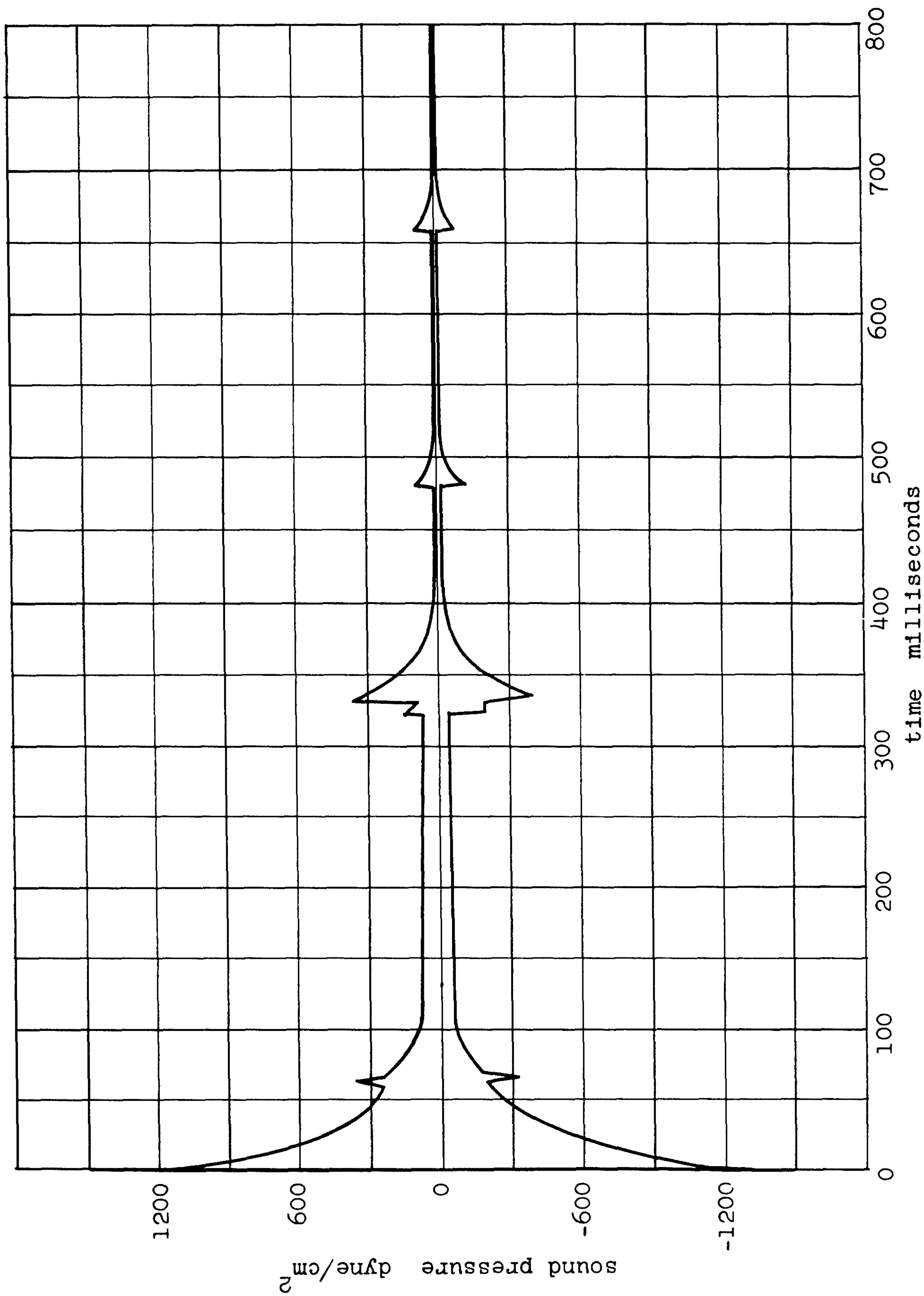

sound pressure dyne/cm^2
1200
600
0
-600
-1200
time milliseconds
0
100
200
300
400
500
600
700
800

Figure 7.10

Sound pressure in dynes/cm^2 vs time for the decay of sound at operator's position after impact of drop forge hammer.

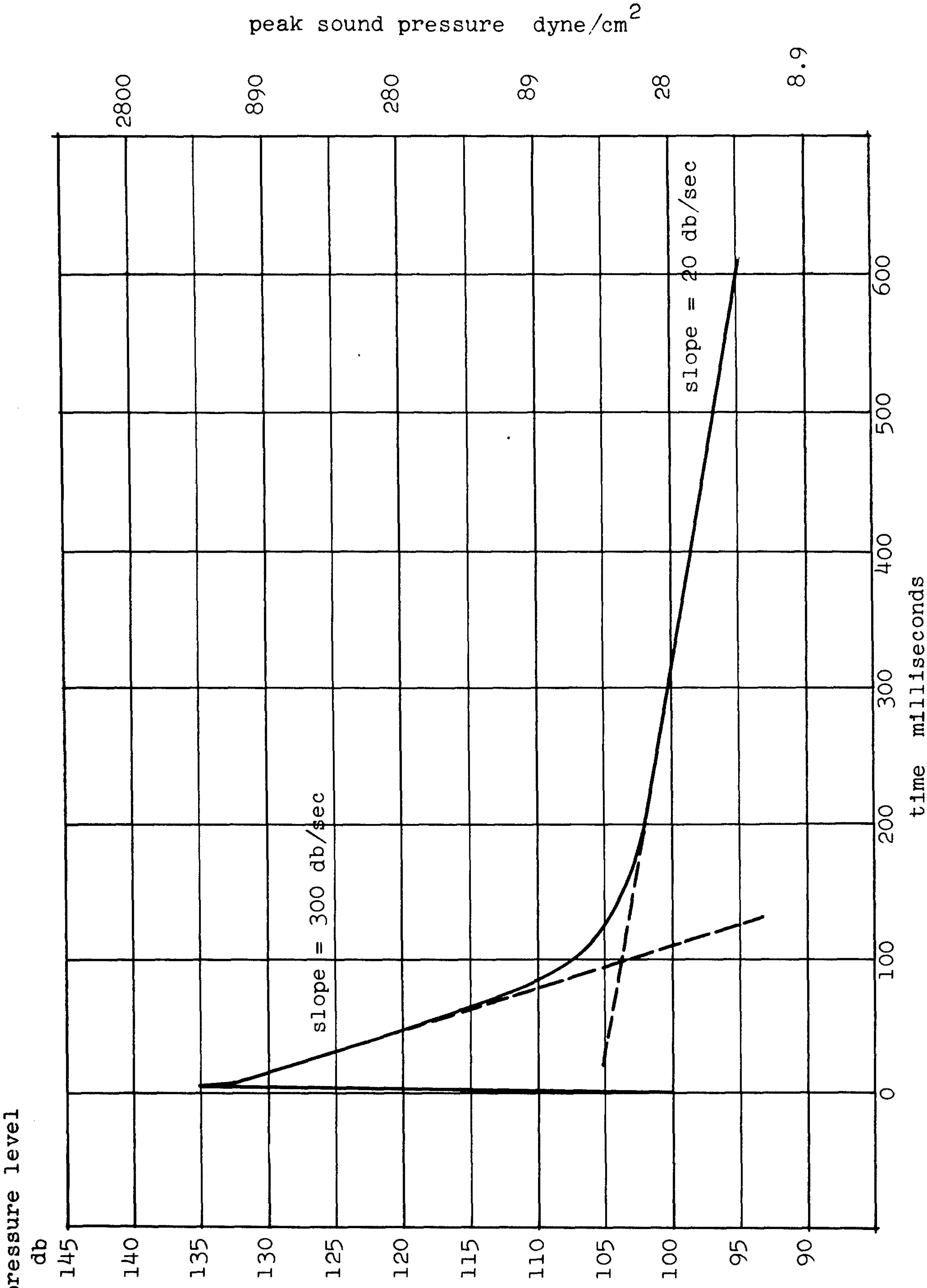

peak sound pressure dyne/cm2
2800
890
280
89
28
8.9
slope = 20 db/sec
slope = 300 db/sec
time milliseconds
0
100
200
300
400
500
600
sound pressure level
db
145
140
135
130
125
120
115
110
105
100
95
90

Figure 7.11

Typical sound decay curve (idealized) for the 800 lb drop forge hammer.

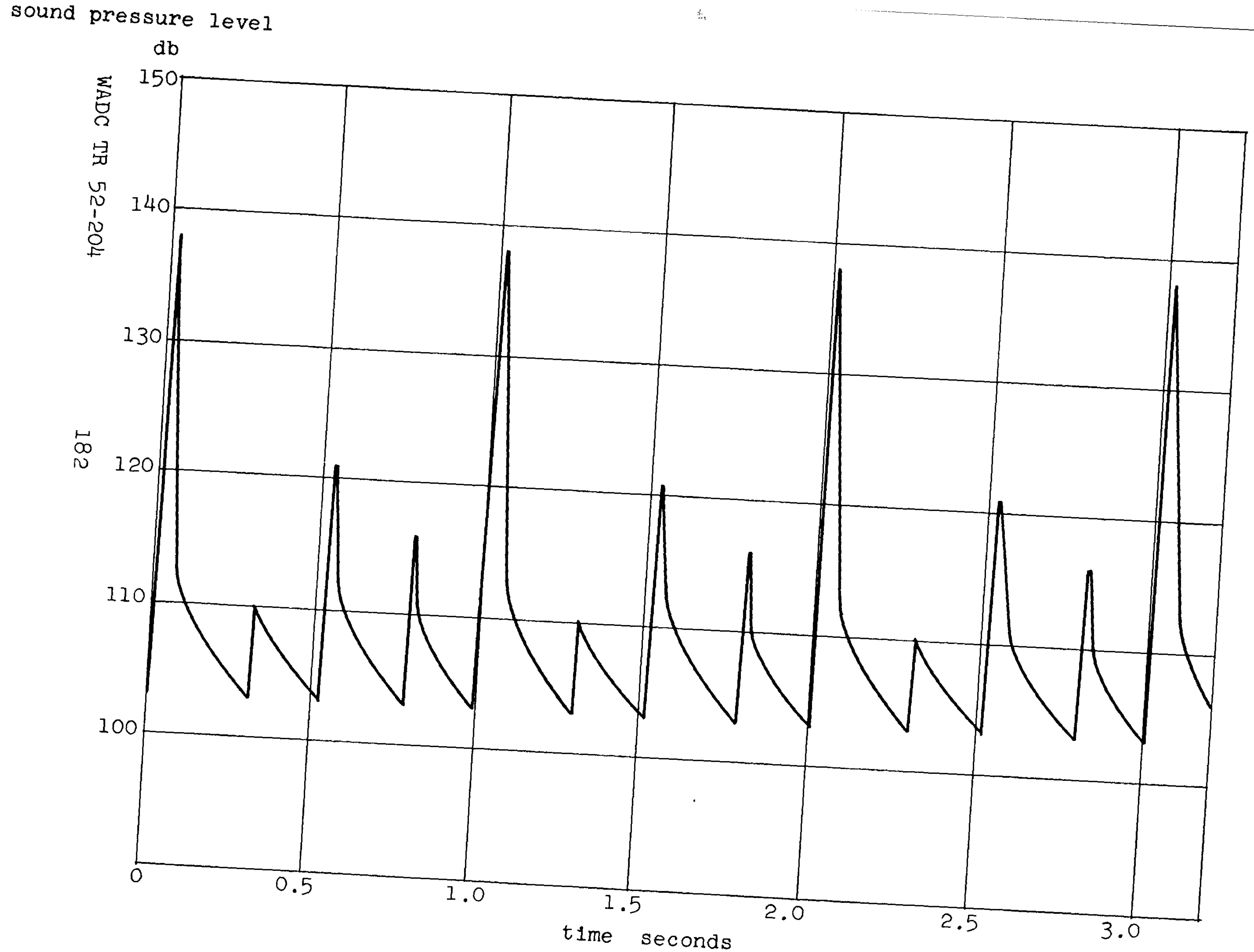

sound pressure level
db
150
140
130
120
110
100
0
0.5
1.0
1.5
2.0
2.5
3.0
time seconds

Figure 7.12

SPL at the operator's position as a function of time when three additional drop forges are operating in the room.

References

(1) This information is based on a survey made by Bolt Beranek and Newman for the Steel Founders' Society of America. A condensed report of the survey was published in Research Report No. 26, Technical Research Committee, S.F.S.A., Sept. 1951.

(2) See Acoustic Measurements, L. L. Beranek, Wiley and Son (1949) pp. 668-678, for a discussion of sound radiation in rooms as influenced by distance and absorption and see Chapter 3 of this handbook.

CHAPTER 8

PHYSICAL CHARACTERISTICS OF MISCELLANEOUS ENVIRONMENTAL NOISE

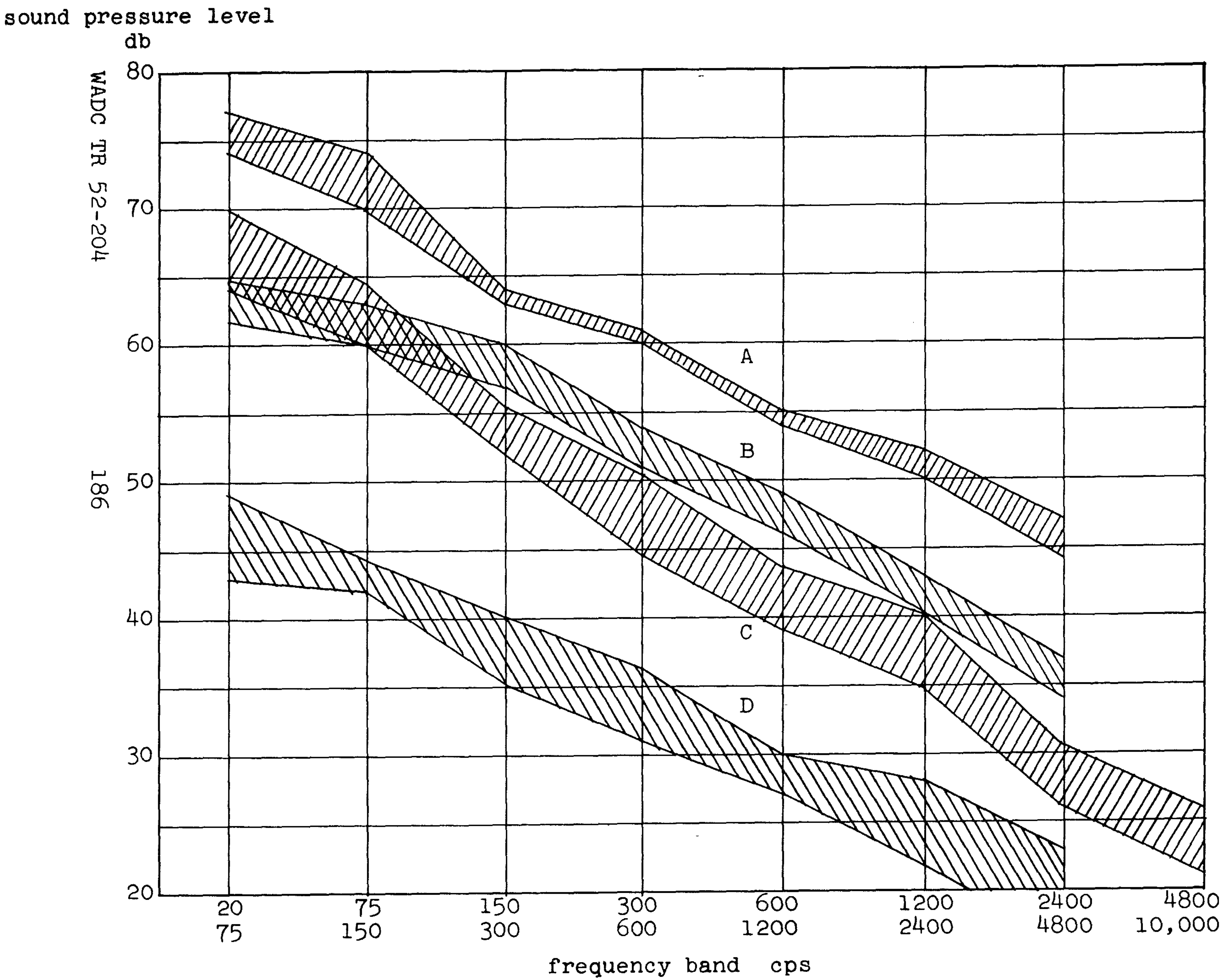
8.1 Introduction

This chapter contains the results of physical measurements of various environmental noises in commercial and residential areas. In order to describe the conditions of the measurements, it is necessary to resort to such general terms as "spacious residential neighborhoods" or "quiet offices". The reported data, however, should be interpreted as a purely physical statement of existing conditions. The terminology which is used is not intended to imply any policy judgements as to the desirability or lack of desirability of any of the situations which are found to exist.

Noise conditions arising in areas where people carry on business activities or where they live may be subdivided into two classifications: (1) Noises which arise primarily outside the work or living space and (2) noises which are associated with the "tools" of an office or a home. Sounds which may be classified as "outdoor" noise affecting offices and homes (particularly in the summertime when windows are normally left open) are first those associated with vehicular traffic. Another type of outdoor noise which may be transmitted into offices and homes is the noise of airplanes. Some information regarding the noise produced by airplanes in flight may be deduced from the data of Chapter 4. Other examples in the present category are noises arising from nearby manufacturing and construction activities, or noises arising from other dwellings.

Office noise sources which fall under the "indoor" classification are, for example, those associated with various types of accounting and secretarial machines such as typewriters, card-sorting and punching machines and teletype machines. Indoor noise sources in the home would include fans, radios, and kitchen appliances such as washing machines, refrigerators, disposal units and food mixing and grinding machines. Only a small amount of relevant data exists concerning home noises. Sample data are included in a later portion of this chapter.

The most usual practice, in dealing with noises falling in the categories of this chapter, is to report actual spectral distributions of SPL at specified locations, rather than to


sound pressure level
db
80
70
60
50
40
30
20
A
B
C
D
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

report the power level spectrum for each of the noise sources. This simplified practice is adopted because most often the observed noise is a superposition of many different sources, operating randomly in time and in some cases at varying positions. A complete study of the individual sources and their times and positions of operation, when many sources are involved, is difficult and is sometimes less useful than a statistical description of their combined noise level. Thus it will appear that a number of the noise levels in this chapter are statistical averages of the effects of many sources which are not individually identified. With a partial exception in the case of traffic noise, the studies have not progressed sufficiently to yield any accurate measures of the statistical variability. In those special cases where the noise arises primarily from a few distinguishable sources to which remedial treatments are to be applied, it becomes important to know the detailed properties of each source. Then the problem must be approached in the manner illustrated in Chapters 4 through 7.

8.2 Outdoor Noise in Commercial and Residential Areas

Surveys of outdoor noise conditions in many types of commercial and residential areas have been made in several different cities in the northeastern and midwestern United States. Considerable differences in the noise levels are found in various individual areas. The noise level seems to be dependent primarily on the location of a home or office with respect to industrial areas and the more or less correlated factors regarding the type of neighborhood, e.g., a tenement area or a suburban residential area.

Figure 8.1 shows octave-band noise levels measured in several cities and in different kinds of neighborhoods. The shaded bands in each curve indicate the spread of the measured data taken over periods varying from several hours to several months duration. The top curve A represents noise levels

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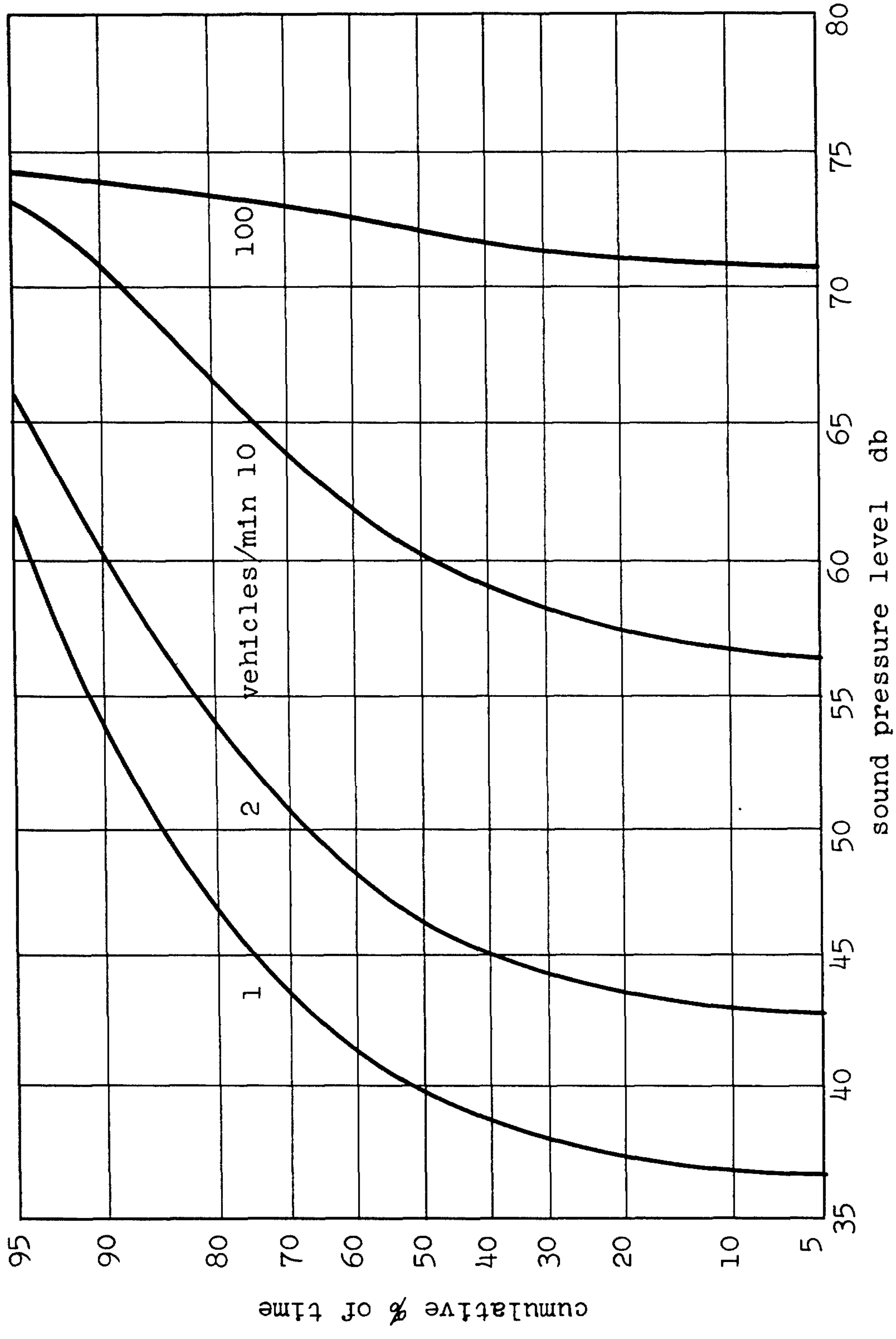
Figure 8.1

Noise spectra in several dwelling areas. Shaded bands indicate spread of observations. A, heavily industrialized waterfront area, "day" and "night"; B, non-industrial neighborhood about 2000 ft distant from heavy manufacturing activity, "day"; C, crowded non-industrial area surrounded by areas of 24-hour industries, "night"; D, spacious residential neighborhoods in several cities, no nearby industrial operations, "night".


100
vehicles/min 10
2
1
35
40
45
50
55
60
65
70
75
80
sound pressure level db
95
90
80
70
60
50
40
30
20
10
5
cumulative % of time

existing in a crowded, heavily industrialized waterfront dwelling area. The traffic consists of buses and heavy trucks. Most of the noise energy is attributable to manufacturing activities. Measurements under both "day" and "night" (after 11 p.m.) conditions are included. The levels vary relatively little during the entire 24-hour period.

Curve B of Fig. 8.1 indicates levels found under "day" conditions in another dwelling area. This area contains single-family houses, each with street frontage of about 60 ft. Heavy passenger car traffic is present. Most of the noise energy is attributable to a heavy manufacturing area about 2000 ft distant.

Curve C shows noise levels for "night" conditions in a large urban residential area which contains no industries but which is surrounded by refineries and manufacturing plants operating on 24-hour schedule. The indicated levels are found, with little variation, along a path crossing the entire area (about 1.5 miles).

Curve D represents "night" noise levels in spacious residential neighborhoods, in several different cities, when no nearby industrial operations are in progress. Under "night" conditions curves B and C drop substantially to the low levels indicated by D on the infrequent occasions when industrial activity ceases in the localities to which those curves apply.

Traffic Noise Levels. Figure 8.2 summarizes the results of noise measurements for uniformly moving traffic. The chart refers to noise levels found at a distance of 20 to 30 ft from a traffic lane used by average passenger cars at speeds from 35 to 45 mph. Since the noise level fluctuates with time, the results are given in terms of the percent of time during which the level will lie below a specified value. The average number of vehicles per minute appears as a parameter. For

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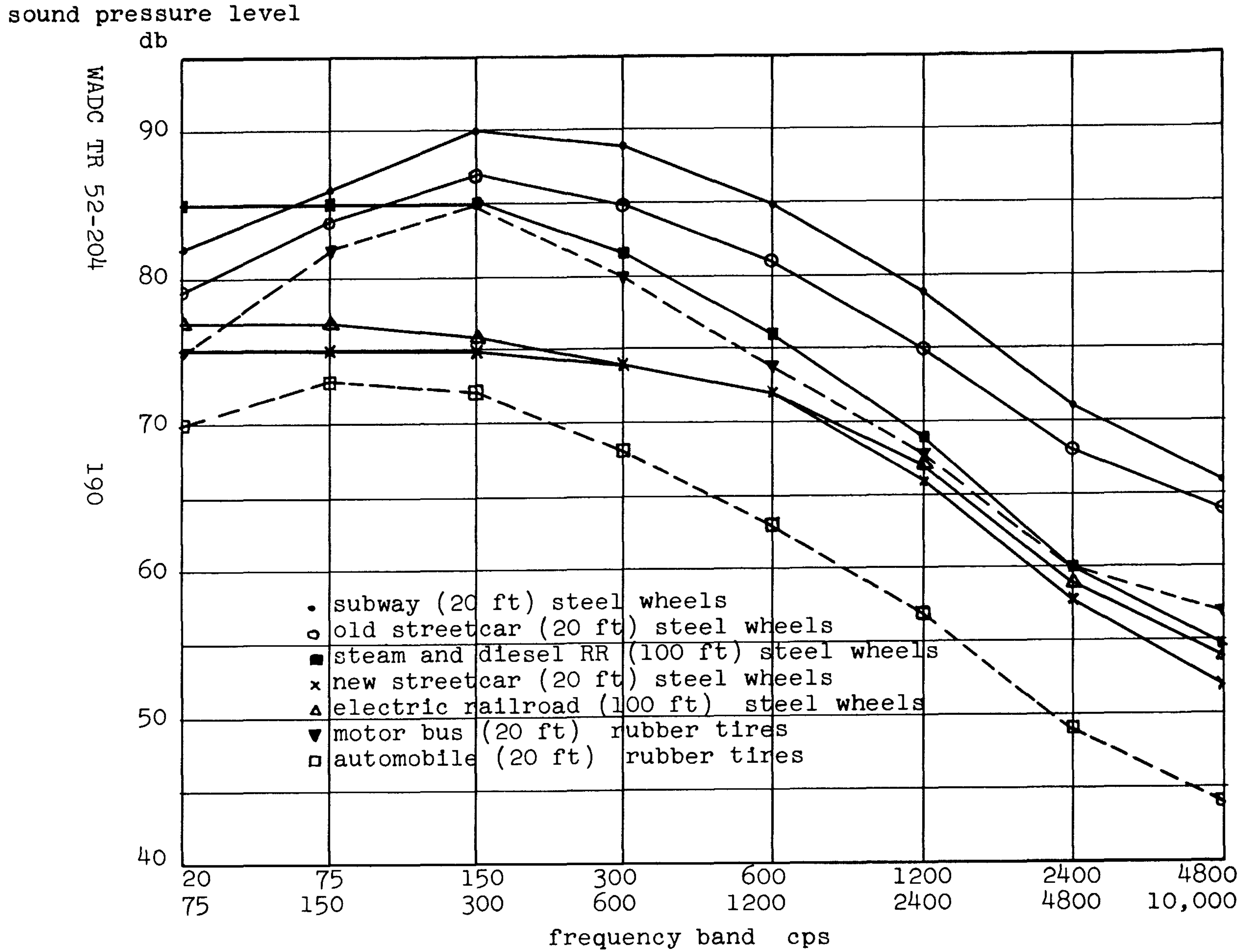
Figure 8.2

Chart, based on traffic noise survey, showing percent of time that overall SPL lies below any arbitrary value, for several specified traffic rates in vehicles/min. Chart applies to observation at 20-30 ft from traffic lane; average passenger cars at 35-45 mph. SPL values should be reduced by 15 db for distance of 150 to 200 ft. Levels for heavy truck traffic are 15 db above passenger car values.


sound pressure level
db
90
80
70
60
50
40
subway (20 ft) steel wheels
old streetcar (20 ft) steel wheels
steam and diesel RR (100 ft) steel wheels
new streetcar (20 ft) steel wheels
electric railroad (100 ft) steel wheels
motor bus (20 ft) rubber tires
automobile (20 ft) rubber tires
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

example, according to the figure an average rate of 10 vehicles per minute would result in an overall SPL which lies below 58 db about 30 percent of the time. The chart terminates at 95 percent. The last few percent of time correspond to the short intervals when the vehicle is very near to the observation point. During these intervals the overall SPL is about 75 db for an average car. When the traffic rate is of the order of 100 vehicles per minute, the level remains near to the maximum value of 75 db at all times.

While the chart applies directly to the case of observations made 20 to 30 ft from the traffic lane, the results can be corrected approximately to a distance of 150 to 200 ft by subtracting 15 db from the indicated levels. If the traffic consists of heavy trucks instead of passenger cars, all sound level values are increased by about 15 db.

The time average of the overall SPL under the above conditions, but with a mixture of truck and passenger car traffic, is given approximately by the summarizing formula

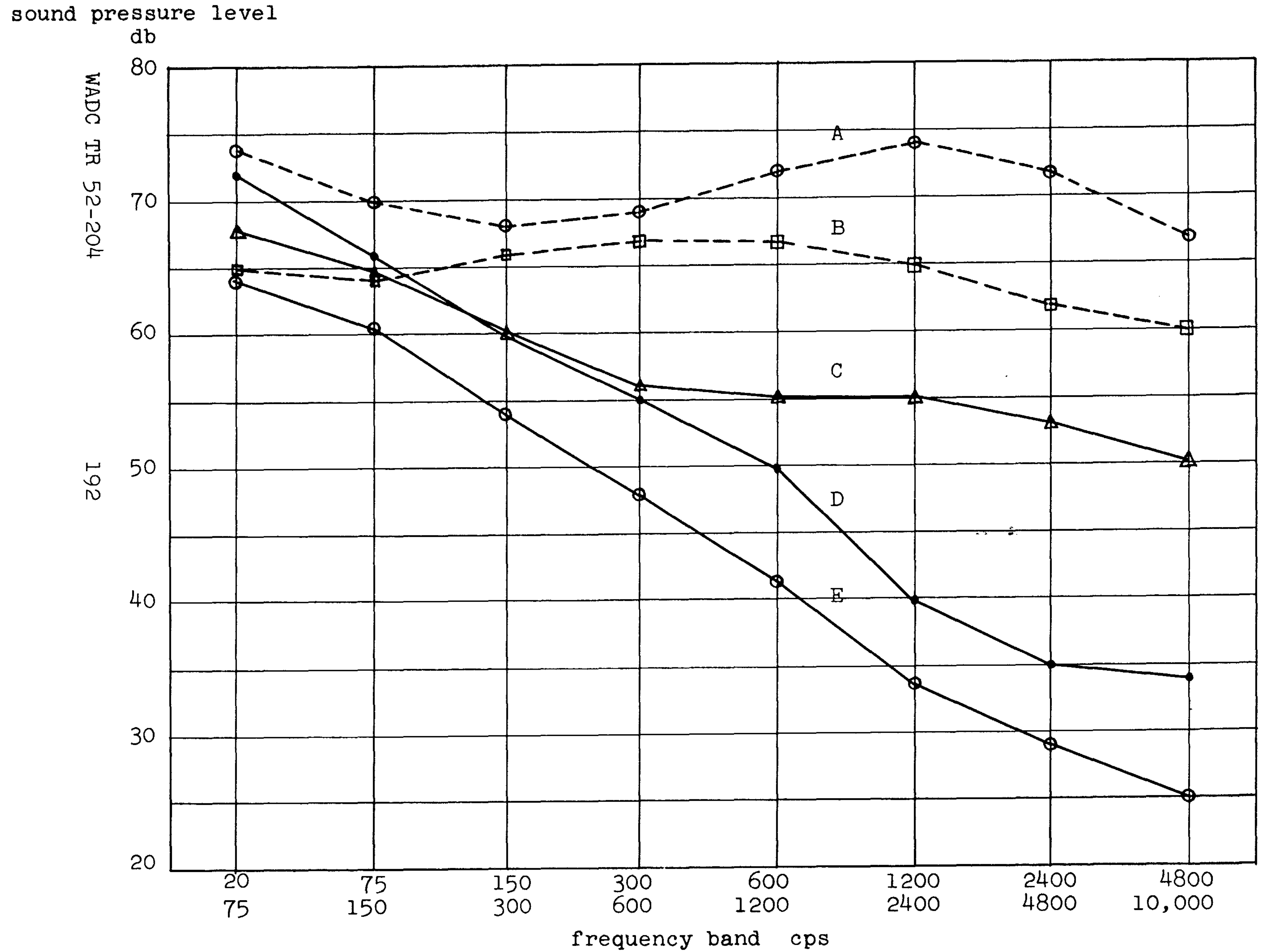
$$\text{SPL} = 83 + 8.5 \log n - 20 \log d \tag{8.1}$$

where _d_ is distance in feet from the traffic lane to point of observation, and _n_ is the number of vehicles per minute. The formula applies to distances of 20 ft or more. Buildings and other reflecting objects are assumed to be absent. It must be emphasized that Eq. (8.1)gives only an approximate effective time average for a fluctuating noise phenomenon. Only when the traffic rate approaches 100 vehicles/min does the noise level at a distance of 20 to 30 ft become constant within a total variation as small as 5 db, even when all vehicles are identical.

Traffic Noise Spectra. Usually the level of traffic noise falls rapidly with increasing frequency. This generalization is apparent from Fig. 8.3, which summarizes noise spectra 1/ for a number of common vehicles at a distance of 20 ft. Other survey information for gasoline vehicles in particular, upon which Fig. 8.2 and Eq. (8.1) are based, leads to the following generalizations which are adequate for ordinary engineering purposes:

Figure 8.3

SPL in octave bands for various vehicles at distance of 20 ft. Data from Ref. (1).


sound pressure level
db
80
70
60
50
40
30
20
A
B
C
D
E
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

1. The traffic noise spectrum diminishes with increasing frequency at a rate of approximately 6 db per octave.

2. The SPL in the 20-75 cps band is about one db below the overall level.

8.3 Indoor Noise Conditions in Offices and Homes

Recent surveys of noise in office spaces indicate that the noise levels in the three frequency bands (600-1200, 1200-2400 and 2400-4800 cps) show a wide variability. This variability can be correlated to some extent with the kind of usage which is made of the office space. In Fig. 8.4 are shown curves representing the noise levels measured in five different kinds of offices. Each curve represents an average of several sets of data measured under the particular conditions indicated for that curve. Curve A shows the noise levels measured in a large tabulating room which has not been acoustically treated. The levels indicated in curve B are for large IBM and teletype rooms. It is well to note that this curve (B) and curve A are for rooms containing similar types of office machines. The disparity in the levels in the high frequency bands may be due to several factors, such as measuring location, size of room, differences in acoustical absorption, and configuration of the room, as well as peculiarities of individual machines.

Curve C was constructed from data taken in a large secretarial office with a constant hum of conversation, approximately 10 typewriters operating, and with one teletype machine operating intermittently. In this office the ceiling was treated acoustically.

The levels shown in curves D and E are for so-called "private" or "executive" offices. In both categories, the rooms had acoustically treated ceilings. The levels in curve D are for those offices rated as "noisy" by their occupants while those of curve E are for "quiet" offices or conference rooms. In offices in both of these groups, telephone conversation

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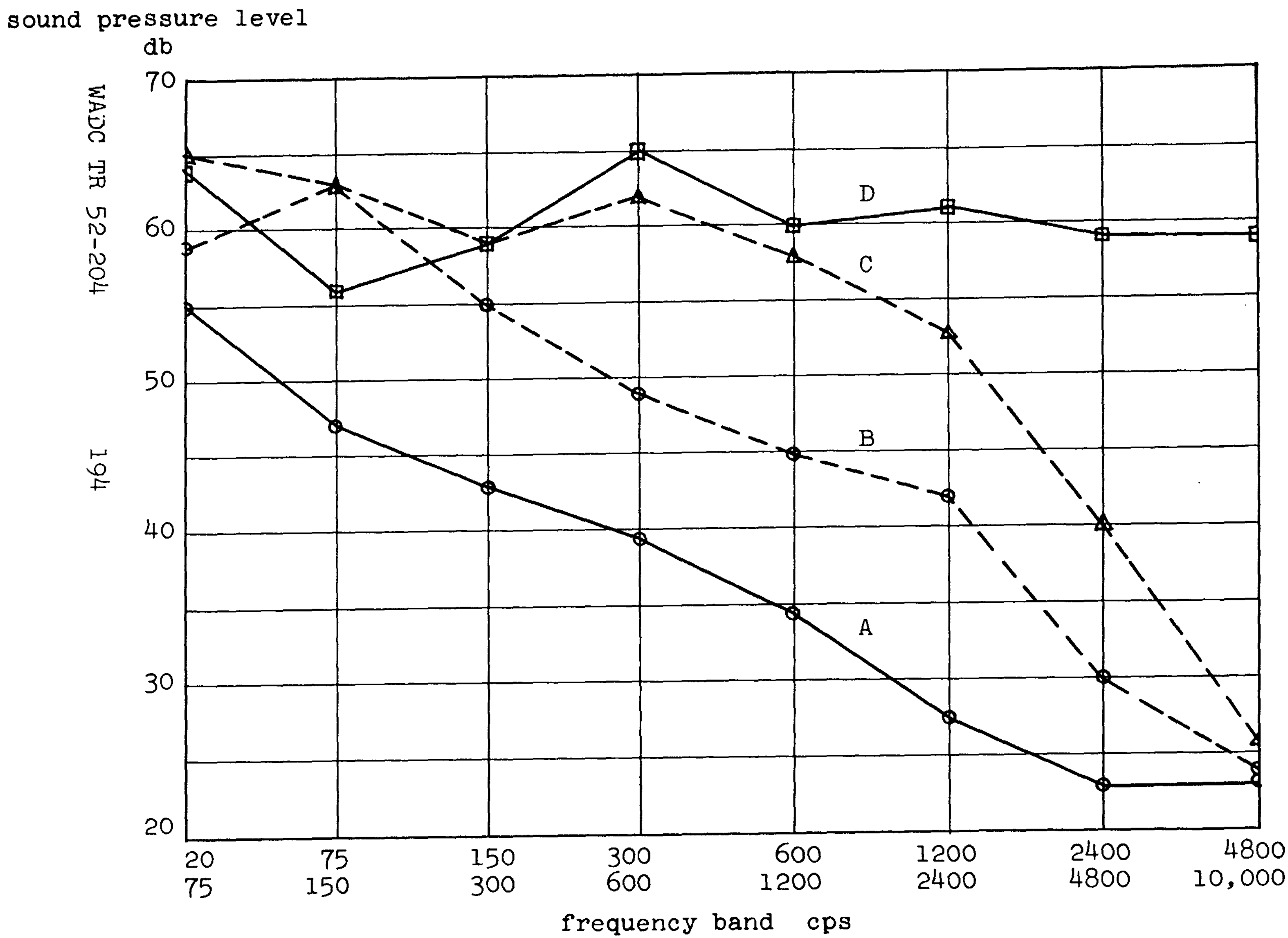
Figure 8.4

SPL spectra obtained in several offices. A, tabulating room; B, large tabulating and teletype room (no treatment); C, 20 ft x 30 ft secretarial office (10 typewriters, one teletype, treated ceiling); D, average of private offices considered "noisy" (treated ceiling); E, average of private offices considered "quiet" (treated ceilings).


sound pressure level
db
70
60
50
40
30
20
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps
D
C
B
A

and normal speaking could be carried on effectively. The "noisy" offices were in every case located close to streets carrying heavy truck and auto traffic, while the "quiet" offices were usually on the second and third floors, away from street noise.

Noise Levels Within Homes. Noise levels within homes can be discussed logically in terms of (1) background levels attributable to external noise, and (2) levels produced by home activities and devices used in the home. Very few measurements of household noise are available. Since there is wide variation as regards both noise sources and acoustic properties of the structure, it is not at present possible to give engineering generalizations. The order of magnitude may be obtained from the spectra for special cases which are plotted in Fig. 8.5. These show background levels and the levels found with various single sound sources functioning within the room, as indicated. The measurements were made with a SLM and OBA, with the microphone placed approximately at the center of the room (15 ft x 15 ft x 9 ft furnished living room, plaster walls, heavily carpeted floors, windows open; residential area one block from traffic artery; "night" condition).

Figure 8.5

SPL spectra under various conditions in a domestic living room (described in Sec. 8.3). "Night" conditions, windows open. A, background due to external noise; B, levels with 10 in., high-speed fan operating at 7 ft from microphone; C, levels with table radio operating at volume considered "normal" for existing background, 7 ft from microphone; D, telephone ringing, 8 ft from microphone.

References

(1) Bonvallet, G. L., "Levels and Spectra of Transportation Vehicle Noise" J. Acoust. Soc. Am. 22 201.

PART III. METHODS OF NOISE CONTROL

CHAPTER 9

GENERAL PLANNING

9.1 Introduction

In Part II the characteristics of a wide variety of noise sources have been discussed. In the following chapters specific principles and techniques of noise control will be presented. Design proposals or specifications regarding the arrangement or construction of buildings, shops, hospitals, offices, test cells and the like can then be derived from an integration of the material in each section relevant to a particular problem. Since there are in many instances a number of differing, but equally valid, techniques available for application to the solution of a given problem of noise control, the designation of the method finally adopted requires the exercise of engineering judgement in evaluating the contributing factors of cost, convenience and practicality. The control of noise radiation from high intensity fixed sources, such as aircraft engines on test stands, is a particularly complex problem requiring close attention to details and the use of every possible expedient for noise reduction. It is in cases such as these, among others, which require that acoustical design considerations must be extended beyond the actual physical structures immediately involved, usually to such as extent as to become a factor in the planning of the base or facility, and its relationship to the adjacent residential or community areas.

In the planning of such specialized facilities as airfields, repair or modification centers, or research establishments including test sites, the initial decision which must be made by management or its governmental counterpart, a central planning agency, is the designation of the general region or area in which it is to be located. Following the top level decision as to general locale based on economic factors or military logistics, the design engineers and architects are confronted with the problem of specific site selection. Considerations which influence this group are usually those geographic and topographic features which will tend to minimize the original cost or difficulty of construction and later operation of the facility. These factors include accessibility of the site by highways or railroad both during and after the construction

period, and location with respect to the nearest utilities (gas, oil, water, electricity, sewer). It is also advantageous to secure a site having a maximum of flat, table-like land.

9.2 Site Selection from an Acoustical Viewpoint

In addition to the above factors, planning engineers must consider the possibility of annoyance to surrounding towns or residential areas due to noise sources which must operate regularly or intermittently in the course of the regular activity of the facility. Once the relative positions of the fixed sources within the physical limits of the base have been tentatively established, it is then possible to calculate on the basis of simple inverse square attenuation of sound with distance, the approximate composite sound levels to be expected at positions outside the immediate area of the base. It is assumed here that estimated or exact information is available regarding the power level and general acoustic characteristics of the sources, under actual operating conditions. From the calculations, the boundaries of approximately concentric regions, each characterized by differing noise level criteria, may be established. In the outermost region the additional noise contributed by the source is negligible; the effective level is the ambient and depends on the nature of the region, either industrial or residential. The next inner region is characterized by a sufficiently higher noise level contribution from the sources so as to be undesirable for suburban residential areas but tolerable for urban residential and industrial areas. Within the innermost region would lie areas having still higher levels and would consequently be unsuited for residential, hospital or educational use and undesirable for most industrial applications. Of course, depending on the number, disposition and treatment of the sources, the higher levels of annoyance may not be experienced; or, conversely, in some cases of very high intensity sources, there may exist an additional inner region where the noise levels constitute a health hazard.

In the setting up of the above "regions" the possible influence of topography associated with the site must not be overlooked. A steep cliff or barrier of sufficient height will often serve effectively as a barrier to the sound radiated by sources located near its base. Topological depressions can sometimes be used to contain a noise source thereby providing an inexpensive sound barrier. The strategic placement of buildings can be employed to form an artificial topography which can be of advantage in some situations. For example, it might be possible

to arrange aircraft jet engine test cells as shown in Fig. 9.1 where a mutual benefit is derived from the isolation walls of opposing cells.

It rarely can be assumed that meteorological conditions are even approximately static. As will be seen in Chapter 12 dynamic variations in meteorological conditions can alter profoundly the assumptions regarding the effects of topography discussed above. The refraction effects which can arise from wind or temperature gradients can produce a considerable reduction in the effective shielding provided by obstacles such as ridges, hills or walls. With sources exhibiting a pronounced

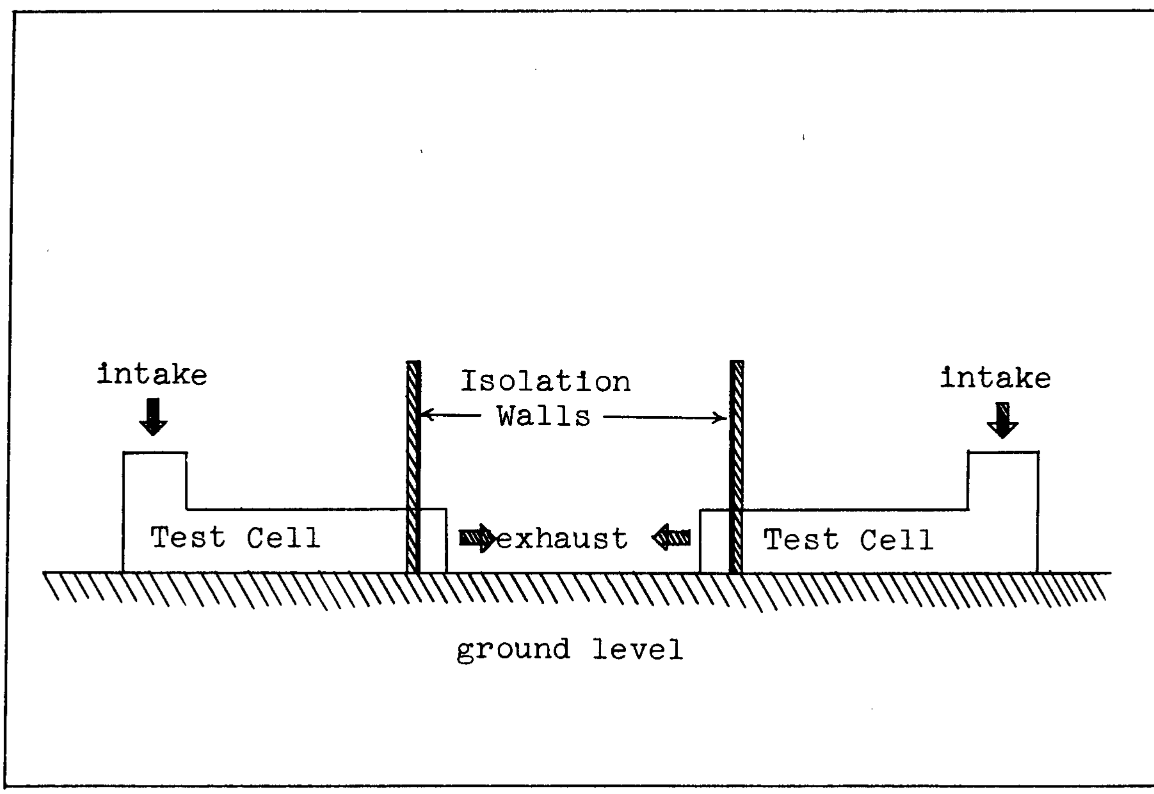


Figure 9.1

Arrangement of aircraft engine test cells to provide mutual shielding by opposed isolation walls.

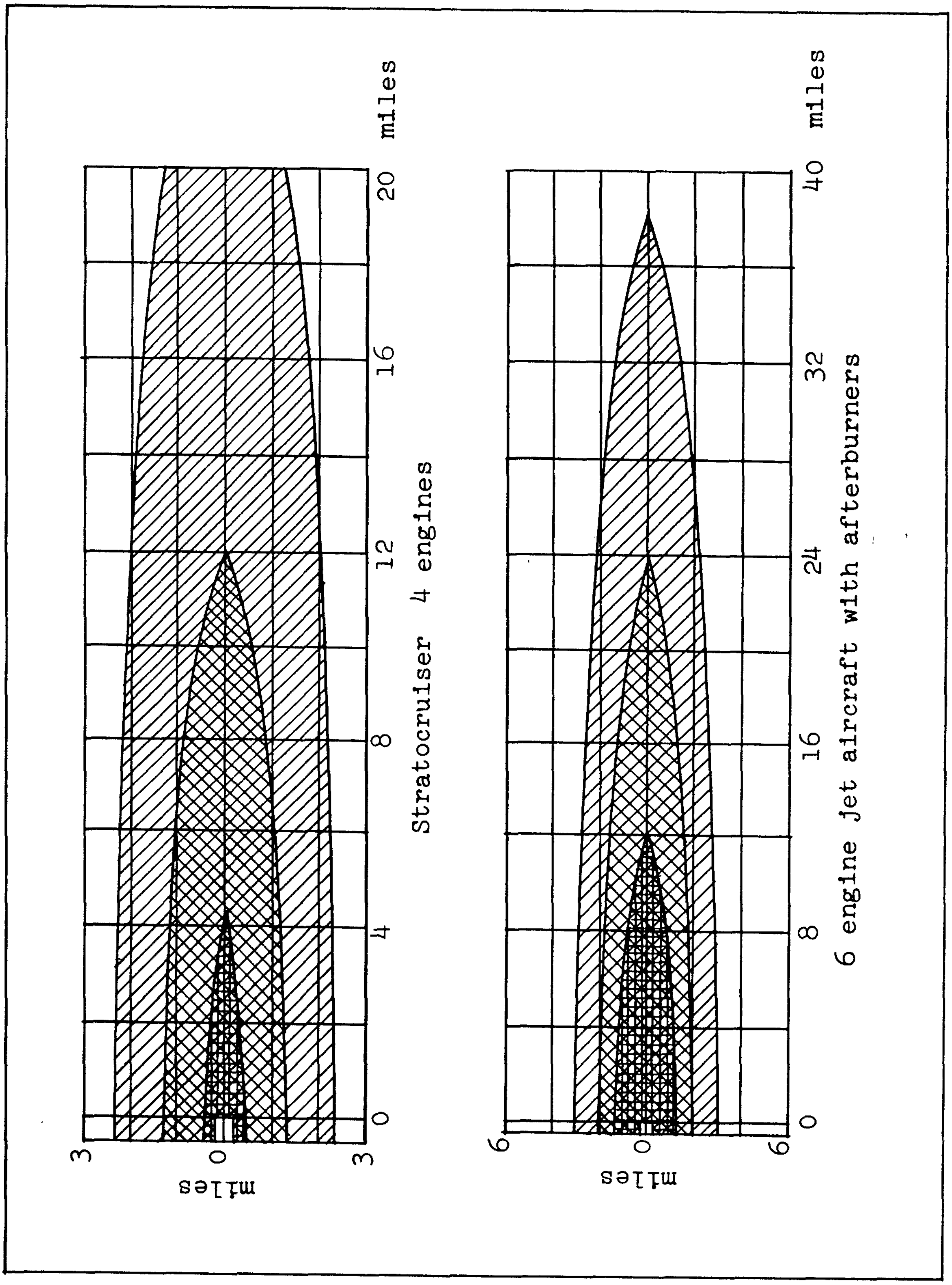
directionality it is often feasible to orient the source so that the principal radiation takes place over unused or waste lands or over a body of water. In such planning, knowledge of the prevailing winds should be used and also the probable effects to be expected when these change. Obviously a meteorological survey of the site is highly useful together with a knowledge of the general weather history of the region.

Additional factors influencing the propagation of air-borne sound include the types of natural absorptive surfaces and foliage that surround the site. A field of tall grass, marsh areas or even snow covered regions will affect the propagation of sound.

The planning and location of airfields involves a number of specialized acoustic considerations in addition to those which have been outlined above. The operation of stationary aircraft on the ground during test or warm-up periods constitutes in itself a difficult problem in noise control. The sources involved are usually of moderate or high power level, operating necessarily out of doors and scattered over a variety of locations, hangar ramps, flight strips, etc., which are distributed over the base.

A more serious problem, however, is that presented by the noise from aircraft in flight and the annoyance to surrounding communities for many miles around due to air traffic radiating from the field. Thus noise control problems which are properly associated with the facility extend far beyond the physical boundaries of the field, along the flight paths for distances of from 25 to 30 miles in some cases. This necessitates estimates of the noise levels to be expected from aircraft in flight on the basis of the take-off paths imposed by runway orientation, prevailing weather conditions and normal traffic flow patterns, in addition to the consideration of the noise level in the surrounding areas due to the fixed sources on the ground. Such estimates also require data on the type of planes to be used, engine ratings and take-off angle.

Examples of the loudness levels to be expected at points on the ground directly below and adjacent to the flight path of aircraft overhead are shown in Fig. 9.2. These show the contours of equal loudness level for two types of aircraft, assuming in each case a 5 degree take-off angle. 1/ The regions between these contours have been designated A, B, C, and D.



Stratocruiser 4 engines
miles
0
4
8
12
16
20
3
0
3
miles
6 engine jet aircraft with afterburners
miles
0
8
16
24
32
40
6
0
6
miles

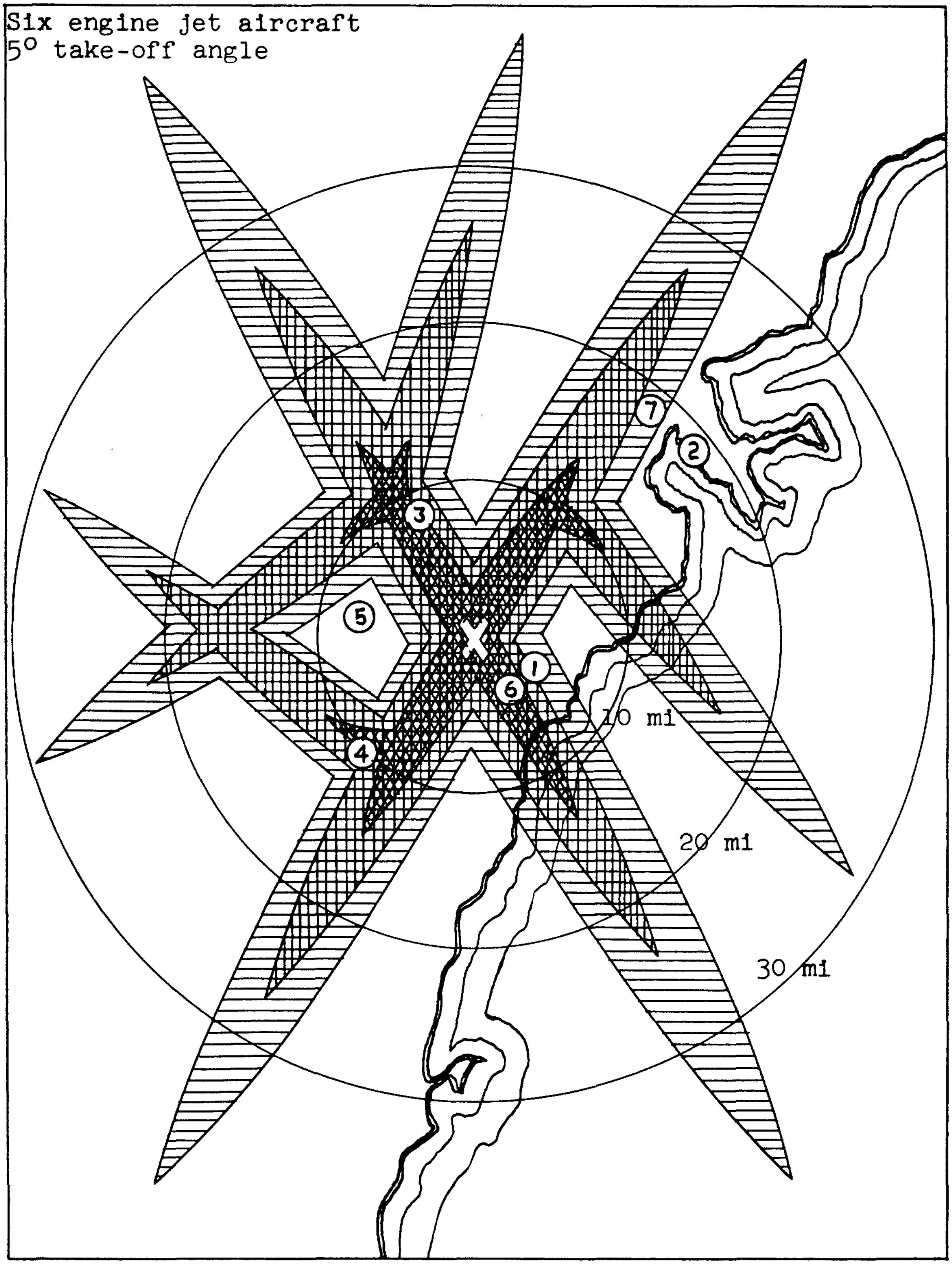
TABLE 9.1

DISTURBING EFFECTS OF AIRCRAFT NOISE

Region	Nuisance (As measured by complaints)	Interference with Speech
A	Negligible (Almost no complaints)	Easy to talk outdoors and indoors
B	Moderate (Occasional complaints)	Necessary to talk in raised voice outdoors. Almost no interference with speech inside frame house.
C	Sizeable (About one-quarter of the population may complain)	Necessary to shout outdoors. Must raise voice inside frame house.
D	Major (Majority will complain)	Impossible to converse outdoors.

Figure 9.2

Regions of major nuisance (black), sizeable nuisance (cross hatched), moderate nuisance (slant-shaded), and no nuisance (white), on the ground beneath the line of flight of two types of aircraft under take-off power conditions at an angle of 5°. The plane rises about 450 ft for each mile indicated on the abscissa. The regions are separated by contours of equal loudness level per band which indicate that on the contour lines the loudness level in each of two or more octave bands is about 40, 60 or 80 phons. These three contours also correspond to speech interference levels of 35, 50 and 65 db respectively. The small white rectangle at the left of the black region indicates one-half of the runway. Four take-offs per hour around the clock are assumed in assessing the nuisance.



Region A
B
C
D

1 Large city
2 Naval shipyard
3 Small city
4 School
5 University
6 Hospital
7 Housing Area

The opinions expressed in the second column are on the basis of several planes per hour taking off. The time when an annoyance occurs is of course very important in determining the subjective reaction. For example, if the noises occur between 9 a.m. and 10 a.m. there will be far fewer complaints from residential areas than if the noises were to occur say between 4 and 5 a.m. The opposite would be true for school or business areas. The duration and rate of occurrence also influences the judgement of these intermittent noises. Combining data such as those shown in Fig. 9.2 computed from measured or estimated data for those aircraft which will form the bulk of air traffic at the base in question, together with the proposed aircraft flight pattern centered on the field, a schematic map such as shown in Fig. 9.3 can be prepared. Here regions A, B, C, and D have been sketched on a map of the proposed field site, together with the location of all major residential areas, towns, cities, schools, hospitals, and industrial areas. It can be established at once from an inspection of such a presentation which areas will be most seriously affected by aircraft noises. It must, however, be recognized that in the area from 10 to 20 miles from the base, deviations from the indicated paths will be frequent depending on the weather, conflicting traffic, etc., and that all points in this region will on occasion possibly experience levels in the C or D range.

Coupled with the subjective annoyance arising from the intrusion by aircraft noise into sleep, conversation or other normal activities is the connotation of potential danger commonly associated with the passage of aircraft overhead. This is not without foundation since analysis of aircraft accidents resulting in casualties to persons on the ground show that these tend to occur on or close to the take-off flight path, on a straight line extended from the airport runway. 2/ For this reason the observer reaction to aircraft noise is frequently apt to be less tolerant than for an intermittent or sustained noise level from a "familiar" source such as street traffic or railroad trains.

Figure 9.3

Typical distribution of aircraft noise levels in the environs of an airfield. This presentation is based on the use of six engine jet aircraft operating from two runway orientations as shown.

9.3 Economics of Planning

The economics of acoustical design and planning is in such a state of evaluation that there is at the present time little definitive information upon which to build a reliable cost index, useful in the initial planning or design stage of a facility. It is possible, however, in certain specific cases to make a comparison of the relative cost of the purchases of land to form protective buffer zones around a noise source as against the cost of building noise attenuating structures. For example, it has been estimated that for an average test cell designed for turbo jet engines the cost of the acoustical treatment required to insure that the noise levels produced at points more than 1/4 mile distant are essentially negligible, may run as high as $90,000.

To this must be added the cost of a structure to house the treatment at an approximate cost of $30,000, giving a maximum cost for the acoustic treatment of $120,000 per cell. These estimates include only those protective measures taken against annoyance to residential areas and does not include the cost of protecting operating personnel or of other incidental noise control measures.

From these figures it is apparent that unless the land within a central region having a 1/4 mile radius (approximately 125 acres) can be purchased for less than $960 per acre or 2 cents per sq ft, the more economical procedure is to arrange some form of acoustical treatment of the noise source. If multiple cell operation (say ten cells) is contemplated, then the cost for acoustical treatment rises to a total of $1,200,000 for the facility and the comparable land purchase rate rises to $9600 per acre or 20 cents per sq ft. Even under these conditions the cost of acoustical treatment may be justified over the cost of purchasing developed residential or industrial areas to form a central region. It is also of interest to consider the savings which might be realized if land outside of a 1/4 mile radius up to 1/2 mile distant were purchased. Buying this additional land will result only in such financial savings as are based on the added sound attenuation effects to be secured by the increased distance to the nearest neighbors. The effective gain in noise reduction, approximately 6 db, will be controlled by that secured at the lower frequencies since it is there that the least benefit from air and surface absorption is secured. The reduction in cost for the required acoustical treatment is about $85 per test

cell per acre of land purchased in the region lying from 1/4 to 1/2 mile from the source. Although these data represent rough estimates they can be used with sufficient accuracy to guide policy discussions at the design stage on the general advisability of purchasing additional land.

9.4 Design of Facility

The need for a complete integration of the various schemes for noise reduction and control cannot be overemphasized. At every stage of planning, the problem of noise intrusion must be related to the decisions as to the arrangement and plant layout as dictated by factors of efficiency, cost and the basic mission of the facility. The need for the planning is present at every level of the design problem and will influence the internal arrangement of rooms within a building, the grouping and relative disposition of buildings within the base area and the orientation and location of the base with respect to the neighboring communities.

In evaluating or proposing various possible building layout arrangements it is well to keep in mind three general building classifications. Buildings which contain such activities as machine shops, foundaries, forges, aircraft test cells, etc., constitute high noise level work areas. They provide in most cases the principal noise sources and may require acoustical treatment to prevent hazard to those working within or near them, to reduce annoyance to other areas on the base, or to prevent annoyance of surrounding community or residential areas. Offices, research laboratories, libraries, schools, etc., constitute working areas which require moderate to low noise levels for efficient operation. Since in many instances these buildings must be located in close proximity to one or more noise sources considerable attention should be paid to building orientation, noise reduction at the source and the provision, as necessary, of additional sound attenuation in the building construction. Rest and recreation areas such as barracks, dormitories, dining halls, and hospitals which require the lowest possible noise levels should be located as far as possible from all noise sources and should be insulated acoustically if necessary.

A number of the more important potential noise sources to be found on USAF bases are presented in Table 9.2 in approximate order of their importance. 3/

TABLE 9.2

POTENTIAL SOURCES OF NOISE - USAF BASES

Flight Activities	Warm-up Areas Approach Zones Maintenance Line Landing Strips
Special Operations and Testing	Engine Test Firing Range Training Areas
Shops	
Transportation	Street Traffic RR Spurs Motor Pool
Recreation Areas	
Utilities	
Food Service	

Most sound sources do not radiate with equal intensity in all directions. While relatively uniform radiation canbe expected at frequencies where the wavelength is much greater than the dimensions of the source, pronounced directionality effects can be often observed at the shorter wavelengths or higher frequencies. This directivity can often be utilized advantageously. In the case of aircraft engine test cells, intake and especially exhaust ducts should be directed toward those regions where the radiation will cause least annoyance. In this connection it is apparent that vertical exhaust ducts, directing the sound upward are often most satisfactory.

9.5 General Aspects of Building Planning

The basic principles which, from an acoustical standpoint, influence the internal arrangement and planning of a building are closely parallel to those already presented in connection with the general planning program for the facility as a whole, i.e., the separation of noise producing areas from those requiring low noise levels, coupled with appropriate measures for noise reduction applied to the sources and the use of sound insulating construction.

It should be emphasized that an analysis of the acoustical and noise annoyance problems present should be made before the planning of a building has proceeded beyond the initial stages. Acoustical treatment or modification of existing buildings are in general more costly and less effective than those appropriate measures which can be taken most properly in the initial stage of design, before the architectural plans are fixed in such details as room size, wall height, wall thickness and types of building material to be used.

Typical noise sources within a building include machine shops, kitchens, offices containing accounting, punched-card or other business machines, heating, ventilating or elevator equipment, and pumps or compressors. Rooms or offices which need quiet surroundings should be arranged on the potentially quieter side of a building, removed as far as possible from noise sources. Storerooms and corridors can often be utilized to serve as isolation spaces between high and low noise level areas. Unless appropriate precautions are taken an air conditioning system can serve both as a source of noise from its machinery, fans and internally produced wind noises, but also to couple acoustically various parts of a building in a highly undesirable fashion.

In the planning and arrangement of the associated duct system, the intake and the exhaust vents, sharp bends should be avoided and in addition as large a piping cross-section area as possible should be employed. In this way noises arising from turbulence and high air stream velocities are minimized. The transmission of noise and vibration along the duct can be reduced or essentially eliminated by suitably designed linings and vibration breaks.

References

(1) Beranek, L. L., "Unsolved Military Noise Problems", J. Acoust. Soc. Am. 24 769-72 (1952).

(2) "The Airport and Its Neighbors", Report of the President's Airport Commission, May 16, 1952, U. S. Govt. Printing Office, Washington, D. C.

(3) Meyer, A. F. Jr., "Sanitary and Industrial Hygiene Engineering Aspects of Master Planning" The Military Surgeon 3 1 (1952).

CHAPTER 10

NOISE CONTROL REQUIREMENTS

10.1 Review of the Quantitative Aspects of Noise-Control Problems

It is the purpose of this chapter to outline the procedures by which acoustical specifications for structural components are derived from the initial information given in a noise-control problem. The discussion will be facilitated by reviewing the quantitative aspects of the problem, as outlined in Chapters 1 and 9. The usual noise-control problem can be summarized under the three headings which follow.

1. Characteristics of the noise source. There is given a sound source producing noise which (at least at certain locations) is undesirable. The quantitative description of the source consists of (a) the power level spectrum, which is a set of power levels for specified frequency bands, and (b) the directivity or directivity index. Ordinarily both the power levels and the directivity values are given for octave-band frequency intervals. In certain problems, for example, those in which the results can be related to the average SPL produced by the source in a reverberant room, knowledge of directivity is not necessary. It is always necessary, however, to know the power level spectrum of the source.

2. There is given a spectrum of maximum allowable values of SPL at one or more specified observation points. The maximum allowable values of SPL are usually derived from bio-acoustic and policy considerations. (The bio-acoustic information is given in Volume II of this manual.) The bio-acoustic data are systematized on the basis of octave bands; therefore, the maximum allowable values of SPL are ordinarily given by octave bands.

3. Specifications for a noise-control system are derived from the data given in (1) and (2). The noise-control system, when used with the sound source having characteristics given in (1), must restrict the SPL at the specified observation points to values not exceeding those given in (2). Since the source data and the maximum allowable SPL values are usually stated according to octave bands, the noise-control specifications are also

usually given by octave bands. Noise-control specifications for discrete frequencies may be introduced when the output of the sound-source is known to consist of discrete frequency components.

The present chapter illustrates quantitatively the principles connected with item (3), that is, with the problem of establishing acoustical specifications for the noise-control system. These principles will be illustrated largely through a numerical example. Further examples which deal with the same basic principles, but with the emphasis on particular techniques for noise reduction, are given in Secs. 12.14 and 13.4. The details of calculating noise reductions due to specific structures are not considered in this chapter, but are given in Chapters 11, 12 and 13.

10.2 Transmission Loss and Noise Reduction

In discussing noise-control specifications, it is necessary to distinguish between transmission loss and noise reduction. The first of these terms is a specific measure of the ability of a particular acoustic component to reduce the transmission of acoustic power; the second is a much more general term, used to describe the effectiveness of any designated combination of noise-control operations in reducing sound pressure level. Noise reduction may include effects due to preferential distribution of sound power (directional radiation) and, if desired, distance effects, in addition to the loss resulting from acoustical attenuating structures. The definitions of the two terms, as given in Chapter 2, are repeated below.

Transmission loss is the ratio, expressed in decibels, of the sound power incident on a sound-control component to the sound power which is transmitted by the component. The term is applied both to building structures (walls, floors, etc.) and to air passages (mufflers, ducts, etc.).

Noise reduction is the decrease, attributable to a designated set of noise-control operations, of the sound pressure level at a specified observation point. Noise reduction is also used to designate the difference in the sound pressure levels existing at two different locations at a single time, when designated sound-control components are in position. A value of noise reduction is significant only when the noise-control components and the points of observation are fully specified.

A specialized form of the term noise reduction, defined below, will also be useful.

The noise reduction relative to an isotropic source is the difference between the SPL which would exist at a designated point if the sound energy diverged uniformly (spherically) from the source, and the SPL which actually exists at that point.

The preceding definition is developed in Chapter 14, which is concerned with the problem of evaluation of a noise-control installation. The concepts of transmission loss and noise reduction are also considered much more extensively in that chapter. It is advisable to study Chapter 14 in conjunction with the present chapter.

10.3 Procedures for Deriving Noise-Control Specifications

1. All pertinent structural and layout information must be assembled. This includes such information as dimensions, wall areas, and volume for an enclosure which is expected to surround the noise source; data for any acoustical absorbing material which is already incorporated in the enclosure design; shape, position, and dimensions of openings; nature of intercommunicating walls; distances from the source to points of observation; any unusualy acoustical data for the surroundings of the observation points.

2. The layout and structures are examined for the purpose of defining all sound transmission paths which may reasonably be regarded as distinct. For example, transmission through the walls of a structure, transmission through the ceiling, and transmission through an opening may be considered to constitute distinct acoustical paths.

3. Each transmission path is considered in detail. Acoustical specifications are assigned to the components (walls, treated ducts, etc.), which insure that the sound power reaching any observation point by this path will make a sufficiently small contribution to the total sound power resulting from all paths.

4. The specifications are readjusted, if necessary, to obtain the least expensive solution to the problem. For example, the total expense might be decreased if the original design for a housing is modified by allowing the walls to radiate more sound power, and by treating the open areas for less radiated power.

In forming the first estimate of the noise control specifications for each path, it is frequently convenient to use as a guide a statement which may be called the principle of equal contributions. This principle is the following: If equal sound powers are delivered by all transmission paths in the absence of acoustical treatment, and if in addition the cost per decibel of noise reduction is the same for all paths, then the most economical noise control is obtained by a design which allows all paths to make equal power contributions to the observed total signal, when the total signal level is adjusted to the design value.

The principle of equal contributions is in no sense a basis for accurate apportioning of sound-control treatment to the various transmission paths in an actual design problem, for in general the initial power contributions of the various transmission paths are not equal, and the costs per decibel of noise reduction for the various paths are not equal. Nevertheless the most economical weighting of the various important transmission paths will usually lie within a few decibels of the equal-contribution condition, because reducing any one power contribution to a value appreciably below the average for all paths will not result in a significant decrease in the total power. For example, if one of two signals of equal power is removed entirely, the SPL drops by only 3 db; if one of four signals of equal power is removed entirely, the SPL drops by only 1.3 db.

10.4 Numerical Example of Deriving Noise-Control Specifications

The numerical example will be developed with reference to the configuration shown in Fig. 10.1, which includes a sound source in an enclosure having one open stack. The sound level developed at a ground observation point, 1200 ft distant, is of particular interest. Since the roof of the enclosure may represent a construction problem distinct from the walls, it is convenient to regard wall transmission and roof transmission as two distinct mechanisms or paths, and to regard the stack as a third transmission path.

The numerical values given below are representative of no actual problem, and are used only for illustration. Certain of the methods of calculation, which will be used with little explanation, are further explained in succeeding chapters.

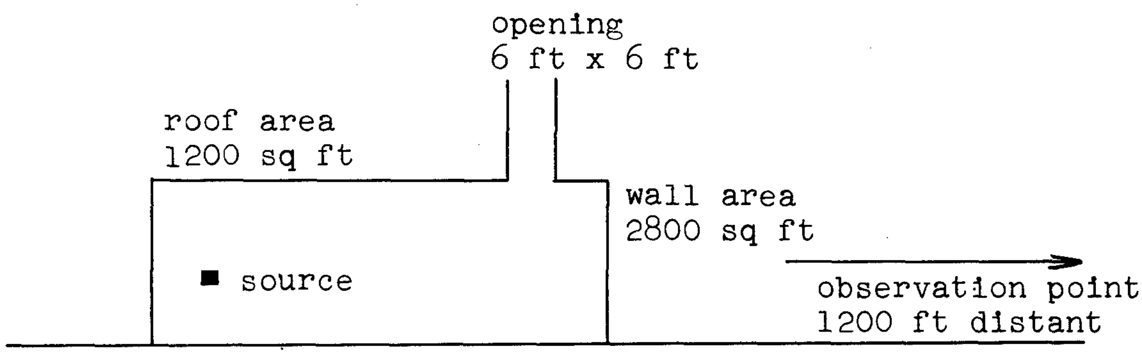


Figure 10.1

Hypothetical example used in deriving noise control specifications. The sound radiated by each part of the structure is evaluated and the transmission loss which must be introduced by that path if certain criteria are to be met is calculated.

The important information concerning the structure and layout is summarized below.

Wall area, 2800 sq ft
Roof area, 1200 sq ft
Stack opening, 36 sq ft
No sound absorbing material within room
Distance to point of observation, 1200 ft.

In addition to the above information, the designer is given the power level spectrum of the source, and the spectrum of maximum acceptable values of SPL at the point of observation. The power level, less ten times the logarithm of the area in sq ft of a sphere of radius 1200 ft, gives the SPL which would exist at the observation point if the source radiated uniformly in free space. (The area factor is thus 10 log ($4\pi \times 1200^2$), or 73 db. The difference between this hypothetical SPL for uniform radiation in free space, and the specified maximum acceptable SPL, is the required value of the noise reduction

relative to an isotropic source. This value of noise reduction does not directly correspond to the acoustical specification for any individual component in the design, but, when properly interpreted, gives a useful initial estimate of the severity of the problem. In particular, if no noise reduction is expected as a result of directional radiation of the sound, then the transmission loss which must be specified for the stack will be equal within several decibels to the noise reduction relative to an isotropic source.

The following tabulation contains the given design values of power level and of maximum acceptable SPL, and shows also the values of SPL which would exist at the observation point in the case of uniform spherical radiation, and the required noise reduction relative to an isotropic source.

Frequency Band, cps	Source power level, db	Max. SPL acceptable at observation pt. db	SPL at observation pt. for uniform spherical rad. db	Required NR rel. to isotropic source db
20-75	163	77	90	13
75-150	164	65	91	26
150-300	161	57	88	31
300-600	158	51	85	34
600-1200	152	46	79	33
1200-2400	146	41	73	32
2400-4800	141	37	68	31
4800-10000	136	33	63	30

For brevity, the remainder of the discussion will refer only to a single frequency band. In a complete set of design calculations, the indicated processes must be carried out separately for each frequency band. Also, in a practical case, some noise reduction will be afforded by directional radiation of sound (Chapter 12), and this should be considered in the design. Since directionality of radiation has not yet been discussed, this feature will be omitted from the present example. The 20-75 cps frequency band has been chosen for the discussion which follows, because directionality of radiation is relatively unimportant in this frequency band for the particular arrangement shown in Fig. 10.1.

The initial specifications will now be derived on the basis that the three transmissions shall make equal contributions to the radiated power. Since the allowable SPL at the observation point is 77 db (in the 20-75 cps band), it follows that the equivalent SPL due to each transmission path alone should be 72 db. (If the total relative power contributed by three equal sources is 77 units on a db scale, we subtract 10 log (3) = 5 db, from which we obtain the relative contribution of each source as 72 units on the same db scale.) In order to allow for possible ground reflections, it will be assumed that the radiated power is distributed over a hemisphere of radius 1200 ft, rather than over a sphere.

Specification for the stack. Since there is negligible sound-absorbing material within the room, the power entering the stack is practically equal to the source power. Then the following relation is valid:

$$SPL_{\text{stack path}} = PWL_o - 10 \log S - TL_{\text{stack}} \qquad (10.1)$$

Here the SPL at the observation point (which must be 72 db) is given in terms of PWL_o, the power level of the source, and S, the area in sq ft of the hemisphere, and TL_{stack}, the transmission loss for the stack. In the present case, 10 log S is 70 db. Thus Eq. (10.1) gives

$$TL_{\text{stack}} = 163 - 70 - 72 = 21 \text{ db}.$$

It will be noticed that this result is 8 db greater than the noise reduction relative to an isotropic source. Of this discrepancy, 3 db is attributable to the allowance for ground reflection, while 5 db is attributable to the requirement that the stack path shall contribute only one-third of the total radiated power. In general, the TL required for a stack is not greater by more than 5 to 10 db than the noise reduction relative to an isotropic source. There is no such simple correspondence in the case of the walls; the TL for a wall may have to be much greater than the initial noise reduction estimate, as the following calculations indicate.

Specifications for the walls. By an area-correction process, the SPL observed on the large hemispherical surface, as a result of wall transmission, may be reduced to the allowable SPL just outside the walls. Thus,

$$SPL_{\substack{\text{outside}\\ \text{walls}}} - SPL_{\substack{\text{obs.}\\ \text{pt.}}} = 10 \log S - 10 \log S_w \qquad (10.2)$$

where S_W is the area in sq ft of the walls. For present data, Eq. (10.2) shows that the allowable SPL outside the walls is

$$SPL_{\substack{outside \\ walls}} = 72 + 70 - 34.5 = 107.5$$

where the number 34.5 is 10 log 2800, the area of the walls being 2800 sq ft.

The SPL inside the room is now calculated. Since the only significant sound absorption in the room is that afforded by the stack opening (36 sq ft of open area, or 36 sabins), the average absorption coefficient for the total of the inside surface is small. The enclosure constitutes a reverberant room. For a reverberant room, the diffuse field SPL is given approximately by the relation

$$SPL_{inside} = 6 + PWL_o - 10 \log A \qquad (10.3)$$

where A is the total absorption in sabins. In the present case, 10 log A is 10 log 36, or 15.5. Then according to Eq. (10.3),

$$SPL_{inside} = 153.5 \text{ db.}$$

The difference between the average inside SPL and the value just outside the walls is known as the noise reduction of the wall structure. For the situation which exists here, where one side of the wall is exposed to a reverberant source enclosure and the other side is exposed to the open air, the noise reduction of the wall structure is 6 db greater than the wall transmission loss (Chapter 11). Therefore, the relationship expressed by Eq. (10.4) is applicable.

$$SPL_{\substack{outside \\ walls}} = SPL_{inside} - (TL_{wall} + 6) \qquad (10.4)$$

When the known values are inserted in Eq. (10.4), the required wall transmission loss is found to be

$$TL_{wall} = 153.5 - 107.5 - 6 = 40 \text{ db.}$$

The procedure used here for finding the required TL for the walls could have been used to find the TL for the stack, instead of the somewhat shorter procedure which was actually given.

Specifications for the roof. The required TL for the roof is found by exactly the procedure which was used for the walls. The only numerical difference arises when the appropriate area for the roof is inserted. The required transmission loss is

$$TL_{roof} = 38 \text{ db}.$$

This completes the derivation of tentative noise-control specifications for the 20-75 cps band, for the simplified situation shown in Fig. 10.1. The designer proceeds to make cost estimates on the basis of the tentative specifications, and then makes any revisions which, by readjustment of the noise reductions for the various paths, will result in reduced cost. Sometimes, on the basis of insight or past experience, the designer can make an approximate initial estimate of the most economical distribution of the noise-reduction expenditures. In this event the first calculations may be carried out on a basis other than that of equal power contributions.

The question might be raised as to why the TL requirements for the walls and roof of an enclosure may be much greater than the noise reduction (relative to an isotropic source) which is afforded by the installation as a whole. The answer is the following: The process of enclosing a sound source within walls, which are not perfectly absorbing, increases the average SPL within the enclosure to values which may greatly exceed the average SPL which would exist in a similar region about the source in the open air. In other words, of the sound power which is incident on a wall having large TL but having a small absorption coefficient, only a small fraction is removed either by transmission or by absorption; most of the incident power is reflected to strike other walls. In the limiting case in which there is no absorption whatever within the enclosure (not realizable in practice), the total sound power will be radiated to the outside, no matter how great the TL of the walls. On the other hand, as the average absorption coefficient for the interior surfaces approaches unity, the wall attenuation becomes fully effective, and the TL requirement for the walls is then of the same order of magnitude as the noise reduction relative to an isotropic source.

CHAPTER 11

CONTROL OF STRUCTURE-BORNE NOISE

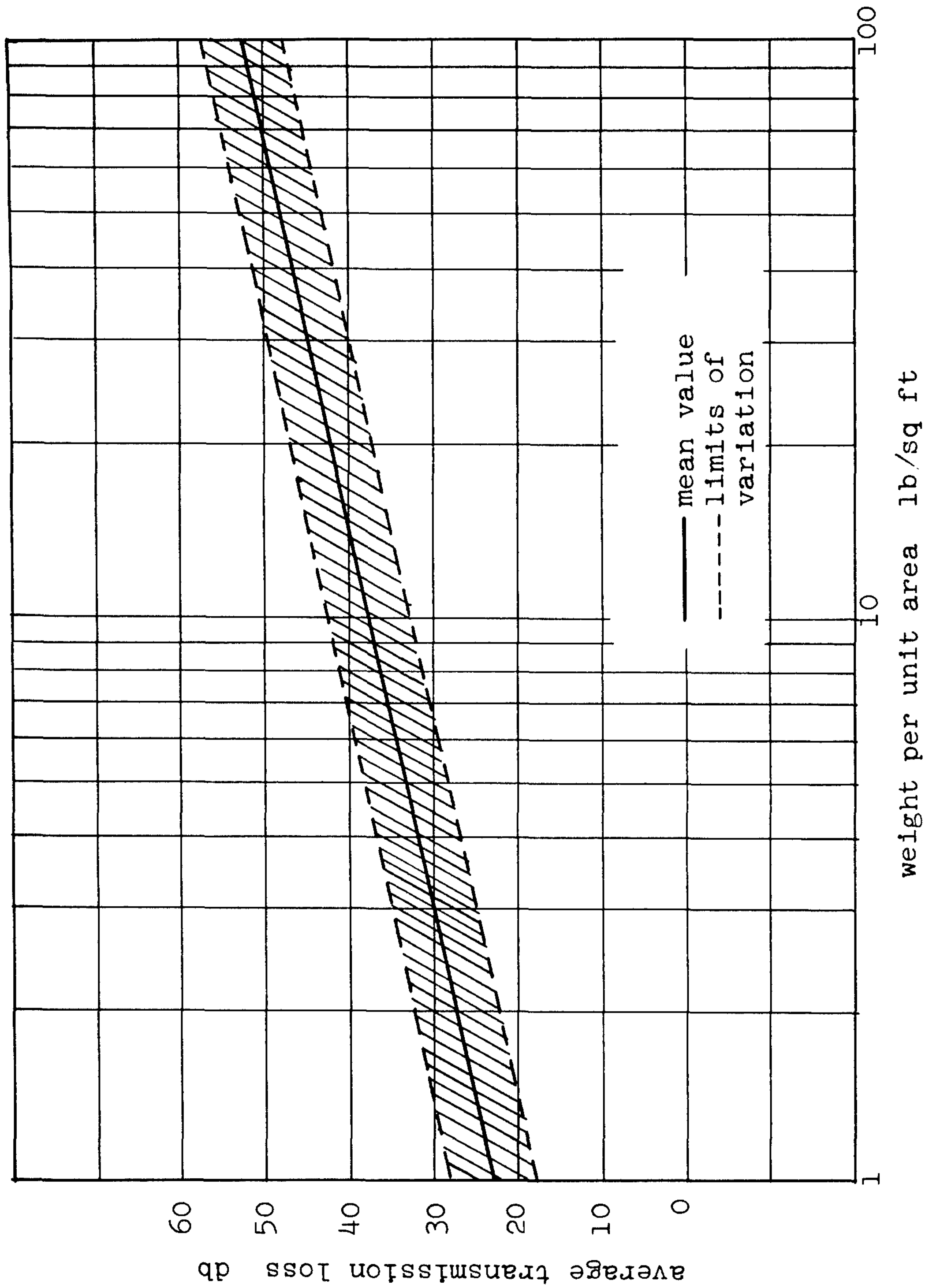
11.1 Introduction

When dealing with sound transmission in buildings it is necessary to consider the problem from two aspects: (1) the transmission of airborne sound, which will be discussed in the next chapter and (2) the transmission of structure-borne sound. In the latter case we are concerned with the problems of the direct transmission of airborne sound through walls and floors and also the radiation, especially from floors, of sound arising from impacts. Since the physical mechanisms involved in these two situations are quite dissimilar, they will be discussed separately in this chapter. This separation is for convenience in discussion. Actual situations will present combinations of most, if not all, of the problems represented and the resulting treatment will consist of a number of noise control techniques.

11.2 Wall Construction

In general the degree of noise insulation provided by a wall increases its surface density. Composite structures, particularly if they include air spaces, are generally superior to homogeneous walls of the same surface density. For example, the sound attenuation provided by two 4 in. brick walls separated by a 4 in. air gap is considerably in excess of that provided by a solid wall 8 in. thick and the attenuation approaches that obtainable with a 12 in. wall. Laminated walls of impervious outer layers, separated by materials of differing density and compressibility (fiber board, soft blanket, light weight fill, etc.) are usually better than homogeneous walls of the same surface density.

Regardless of surface density, however, a wall must be highly impervious to air flow if it is to give a high degree of sound insulation. For example, ordinary walls of unplastered clay tile or porous cinder block provide little sound isolation and are inadequate for inter-office privacy, while the same wall, tightly sealed with plaster or a rubber base or plastic paint on each side will provide good sound isolation. The paint accomplishes a sealing action by bridging the pores in the block with a thin membrane. Water base paints and most oil base paints are unsatisfactory for this purpose.


average transmission loss db
0
10
20
30
40
50
60
weight per unit area lb/sq ft
1
10
100
mean value
limits of variation

The attenuation of incident airborne sound by walls and floors has been the subject of extensive investigation and measurement. Accordingly, there are tabulations of data available in the literature on the performance characteristics of the various types of materials and varieties of construction in common use for walls, doors and windows. 1,2/ These results are commonly expressed in terms of the transmission loss (TL) given by:

$$TL = 10 \log_{10} (1/\tau) \quad \text{decibels} \qquad (11.1)$$

where τ is the fraction of the incident sound energy transmitted by the wall. Values of TL or τ are usually measured and specified at several frequencies. A common set of frequencies is 128, 256, 512, 1024, 2048 and 4096 cps. It is sometimes satisfactory to consider only an average value of TL and its effect on the average sound level. The frequency ranges used in averaging TL values are not standardized. Many references give average values for "low frequencies" and "high frequencies" while others divide the frequency range into three bands.

In Fig. 11.1 several types of walls, floors and panels are illustrated and are arranged according to their transmission loss values. In general, the TL increases as the barrier becomes heavier and more complex. Unfortunately the cost also increases, so that economic compromises often become necessary. When high insulation is required, a variety of possible alternate solutions should be explored. A homogeneous wall increases in TL as its surface density increases. The average trend is shown in Fig. 11.2; the solid line gives the mean value and the dashed lines give the range within which most homogeneous walls lie. The slope is about 5 db increase for each doubling of weight. Above about 50 db the single wall provides a diminishing return and it becomes more economical to split the wall into two or more independent layers. Table 11.1 lists typical values of surface density for a number of common building materials.

From an inspection of Fig. 11.1 it can be seen that splitting a plastered 8 in. cinder block into two 4 in. leaves separated by a 4 in. air space gives a TL of about 55 db which is about 8 db more than would be obtained from a solid plastered cinder block of the same surface density. Even if total thickness,

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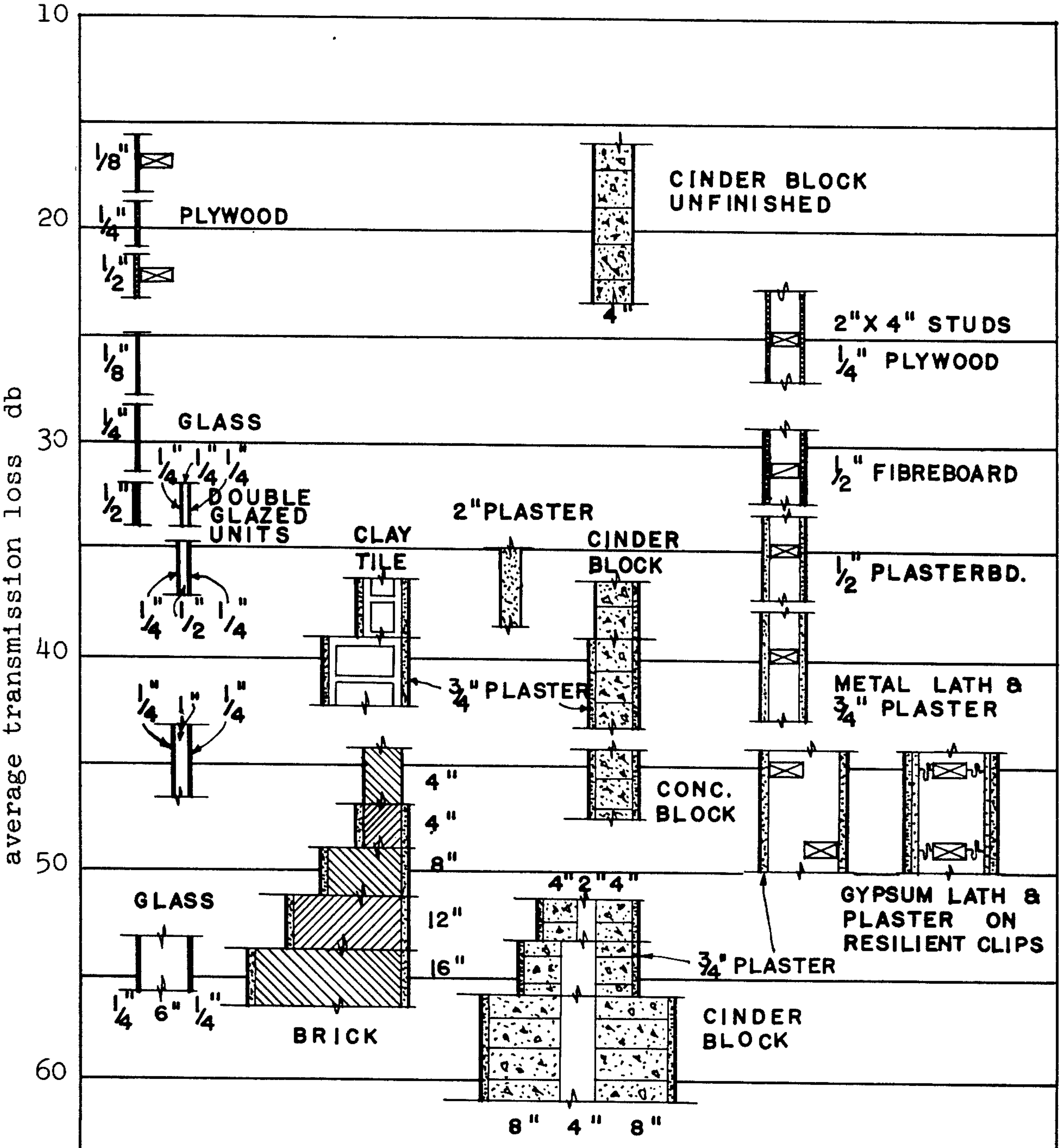
Figure 11.1

Average transmission loss for typical wall construction.



average transmission loss db
10
20
30
40
50
60
1/8"
1/4"
PLYWOOD
1/2"
CINDER BLOCK
UNFINISHED
4"
2"X 4" STUDS
1/4" PLYWOOD
1/8"
1/4"
GLASS
1/2"
1/4" 1/4" 1/4"
DOUBLE
GLAZED
UNITS
1/4" 1/2" 1/4"
1/2" FIBREBOARD
2"PLASTER
CLAY
TILE
CINDER
BLOCK
1/2" PLASTERBD.
3/4" PLASTER
METAL LATH &
3/4" PLASTER
1/4" 1" 1/4"
4"
4"
CONC.
BLOCK
8"
12"
16"
GLASS
1/4" 6" 1/4"
BRICK
4" 2" 4"
3/4" PLASTER
GYPSUM LATH &
PLASTER ON
RESILIENT CLIPS
CINDER
BLOCK
8" 4" 8"

TABLE 11.1

WEIGHT OF COMMON BUILDING MATERIALS IN
LBS PER SQ FT PER IN. OF THICKNESS

Brick	10-12	Lead	65
Cinder Concrete	8	Aluminum	14
Dense Concrete	12	Steel	40
Wood	2-4	Gypsum	5
Glass	15		

instead of total weight, is the imposed limitation it will usually be possible to get higher TL in a split than in a single one.

Stud constructions with wood or plaster facings have similar properties. As long as the two faces are rigidly tied together, the TL follows the general weight law, illustrated by Fig. 11.2. But if the two faces are structurally separated, on separate rows of studs, the TL increases beyond that expected from density considerations alone.

Of practical significance in actual problems of noise control is the calculation of the sound pressure level within an enclosure due to an external noise source, or the determination of the sound pressure level at some point outside of an enclosed source. As will be shown, the computation of these levels requires a knowledge of both the transmission loss of the walls and other structures involved in the direct transmission path, and the total number of effective units of sound absorption in the receiving room or region. Usually satisfactory calculations of the sound pressure level can be made by classifying the source region and the receiving region as either acoustically "live" or "dead". In a "live" enclosure, multiple reflections of sound energy from the walls produce an effectively diffuse sound energy distribution throughout the region. The absence of such reflections, either due to the complete lack of boundary surfaces or the treatment of the walls so as to make them highly absorbent,

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Figure 11.2

Average transmission loss of solid partitions as a function of the weight per unit area of the partition.

results in an acoustically "dead" region.

In the following discussion it will be assumed that the only significant path for sound transmission between region #1, which contains the source, and region #2, which contains the receiver or observer, is that provided by a wall having a total area of S sq ft. This basic presentation is generalized in a succeeding section to include a consideration of multiple paths and sources.

Where both source and receiver are in regions which are "live" the relationship between SPL_1, the sound pressure level in region #1 containing the noise source, and SPL_2, the sound pressure level in the receiving region #2, is given by the relationship

$$SPL_2 = SPL_1 - TL + 10 \log_{10} (S/a_2) \tag{11.2}$$

where

S = the area of the common wall in sq ft
TL = the transmission loss of the common wall
a_2 = net effective acoustical absorption (in sq ft) in the receiving room.

The quantity a_2 can be computed from the individual absorption coefficients of the materials covering the interior surface of the receiving room. Thus

$$a_2 = A \ln [1/(1 - \bar{\alpha}_2] , \tag{11.3}$$

$$\bar{\alpha}_2 = \frac{1}{A} \sum_i A_i \alpha_i , \tag{11.4}$$

where:

α_i = the acoustical absorption coefficient of material "i"
A_i = the surface area of material "i"
A = total surface area of receiving room = $\sum_i A_i$.

Frequently the common wall between regions #1 and #2 is composed of sections having differing values of TL. This will be true, for example, if the wall includes windows or doors. In this case an effective transmission loss $(TL)_e$ must be computed from the tabulated values of TL values for the component sections. For example, consider a wall which includes a door, area S_1, and a window, area S_2. The wall surface remaining has an area S_3 such that $S = S_1 + S_2 + S_3$. The fractional transmission of each

of these materials is first calculated from the tabulated values of TL_1, TL_2, TL_3 using the equation

$$1/\mathcal{T}_1 = \text{antilog}_{10}\,(TL_1/10) \quad \text{etc.} \tag{11.5}$$

then $$\mathcal{T}_e S = \mathcal{T}_1 S_1 + \mathcal{T}_2 S_2 + \mathcal{T}_3 S_3 \tag{11.6}$$

where $\mathcal{T}_e$ = the effective fractional transmission of the common wall.

Dividing gives

$$\mathcal{T}_e = \frac{\mathcal{T}_1 S_1 + \mathcal{T}_2 S_2 + \mathcal{T}_3 S_3}{S} \tag{11.7}$$

hence $$(TL)_e = 10 \log_{10} (1/\mathcal{T}_e) \tag{11.8}$$

If the source region is "live" but the receiving region is "dead", then the sound pressure level measured near the surface of the common wall on the receiving side is given by the expression

$$SPL_2 = SPL_1 - TL - 6 \tag{11.9}$$

These conditions are typical of the situation present when calculating the effect of an enclosure in reducing the outdoor sound level. It is first necessary to evaluate the rate at which acoustical energy is transmitted through the enclosure walls. These radiating surfaces are then considered as a source and have new characteristics determined by the various attenuation properties of the walls, their size and orientation. Calculations of this sort are described in a later section.

A noise source located in a region that is "dead" can nevertheless produce an energy flux at the common wall which is distributed over a range of angles. This will be true when the sound source subtends a large solid angle at the wall or when the wall is sufficiently far from the source to receive the singly reflected waves which will always be present regardless of the amount of absorption on the walls, ceiling and floor. However, the amount of energy arriving at the wall from these reflections is quite small. In such a case, the sound pressure level SPL_2 in an adjacent "live" receiving region is given by the equation:

$$SPL_2 = SPL_1 - TL + 10 \log_{10} S/a_2 + 3 \qquad (11.10)$$

where S and a_2 are defined and calculated in connection with Eq. (11.2).

If both the source and receiving regions are "dead" then the sound pressure level in the receiving room, measured near the surface of the common wall is given by the expression

$$SPL_2 = SPL_1 - TL - 3 \qquad (11.11)$$

For the same value of SPL_1, SPL_2 will be approximately 3 db less than the values computed in Eqs. (11.10) or (11.11) if the sound energy incident on the source side of the wall is a plane, progressive wave.

The preceding equations apply to most situations where it is desired to determine the noise reduction to be expected from existing structures, or to act as a guide in the selection of means to provide adequate sound insulation under specified source conditions. However, in many instances the analysis must itself include a prediction of the probable values of SPL_1, based on the acoustic power output to be expected from the sources to be present. The resulting steady state sound pressure level within an enclosure which is supplied acoustic energy at the rate of P watts depends on the amount of absorption present within the volume a_1 and is given by the expression.

$$SPL_1 = 10 \log P/10^{-13} - 10 \log_{10} a_1 + 6 \qquad (11.12)$$

where the quantity a_1 is computed according to Eqs. (11.3) and (11.4). Note that this equation assumes an essentially "live" source enclosure. Substituting this relationship into the previous Eqs. (11.2) and (11.9) permits the receiving room levels to be calculated directly from the power level of the source.

The design of sound insulating enclosures involves more than a choice of wall structures with adequate transmission loss. Attention must be paid to proper detailing of all joints, divisions, and inserted elements. The acoustic insulation of an elaborate multiple wall can be seriously reduced by air leaks such as occur around doors, windows, and through ventilating openings. An ordinary flush door with a threshold crack has a negligible transmission loss (perhaps 10 db or less) while with a tight rubber or molded neoprene gasket all around, the TL can be 25 or 30 db, a value which provides adequate privacy for most cases. It should be noted here that the noise reduction capa-

bilities of a door or window need not be quite as high as that required for the wall as a whole since these portions usually occupy only a small fraction of the total area. It is very difficult to obtain a TL of more than 35 to 40 db for a single door even with such elaborate precautions as gasket seals and tight fitting hardware. Where there is need for higher insulation at points of entry, double doors or enclosed sound locks, such as are used in broadcasting studios must be employed.

An effect, equally as important as air leaks in determining the degree of sound insulation attainable with a given enclosure, is the transmission of sound energy along structural paths which by-pass the walls. The situation which is usually present is shown in Fig. 11.3 where we have an enclosed source room adjacent to a receiving room with a single common wall between them. As indicated in the figure there are five possible paths whereby sound energy arising in the source room can be transmitted to the receiving room. Airborne sound striking any wall tends to impart at least a small amount of bending vibration to that wall which in turn is communicated to adjacent walls and floors. This structural vibration propagates readily through a continuous rigid frame of steel or concrete, less efficiently through non-homogeneous masonry or wood frame construction, and is blocked almost entirely by structural breaks which are sealed by soft mastic.

Techniques for determining the sound transmission via path 1 have been discussed in detail. Transmission along path 5 can be computed either from measurements or calculation of the actual sound pressure levels immediately outside the receiving room walls. Paths 2, 3, and 4 represent so-called "flanking" effects which obviously cannot be included in the general specifications of the TL of wall materials. While unimportant for cases where only moderate amounts of sound insulation are required, these indirect paths set the upper limit to the isolation that can be achieved between the two rooms, regardless of the effectiveness of the wall between the two rooms. This limit is usually 50 to 55 db depending on the construction. For insulation above this, special detailing must be used to break up these indirect transmission paths.

Situations which require high values of transmission loss may necessitate, in addition to special attention to flanking effects, wall and floor structures of uncommon thickness and weight. In these cases it is well to consider the use of multiple walls of special construction. Four wall types which provide an especially high degree of sound insulation are shown in Fig. 11.4 together with their TL

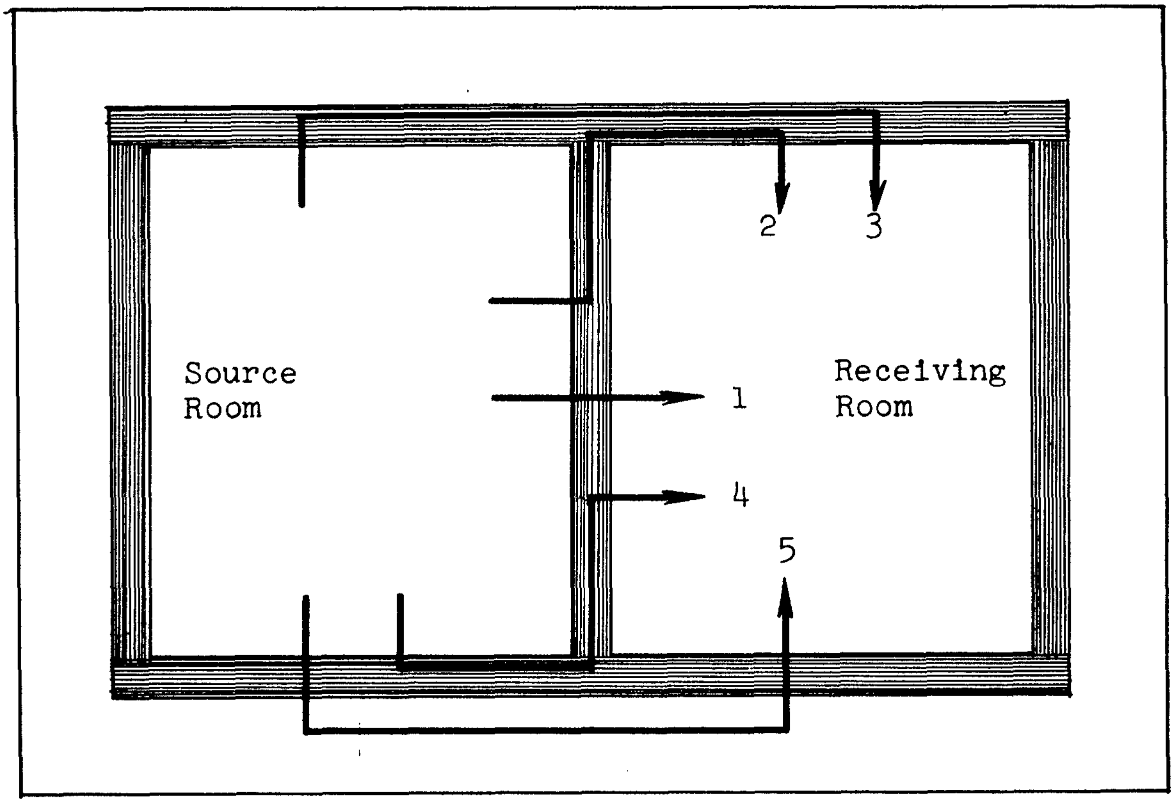
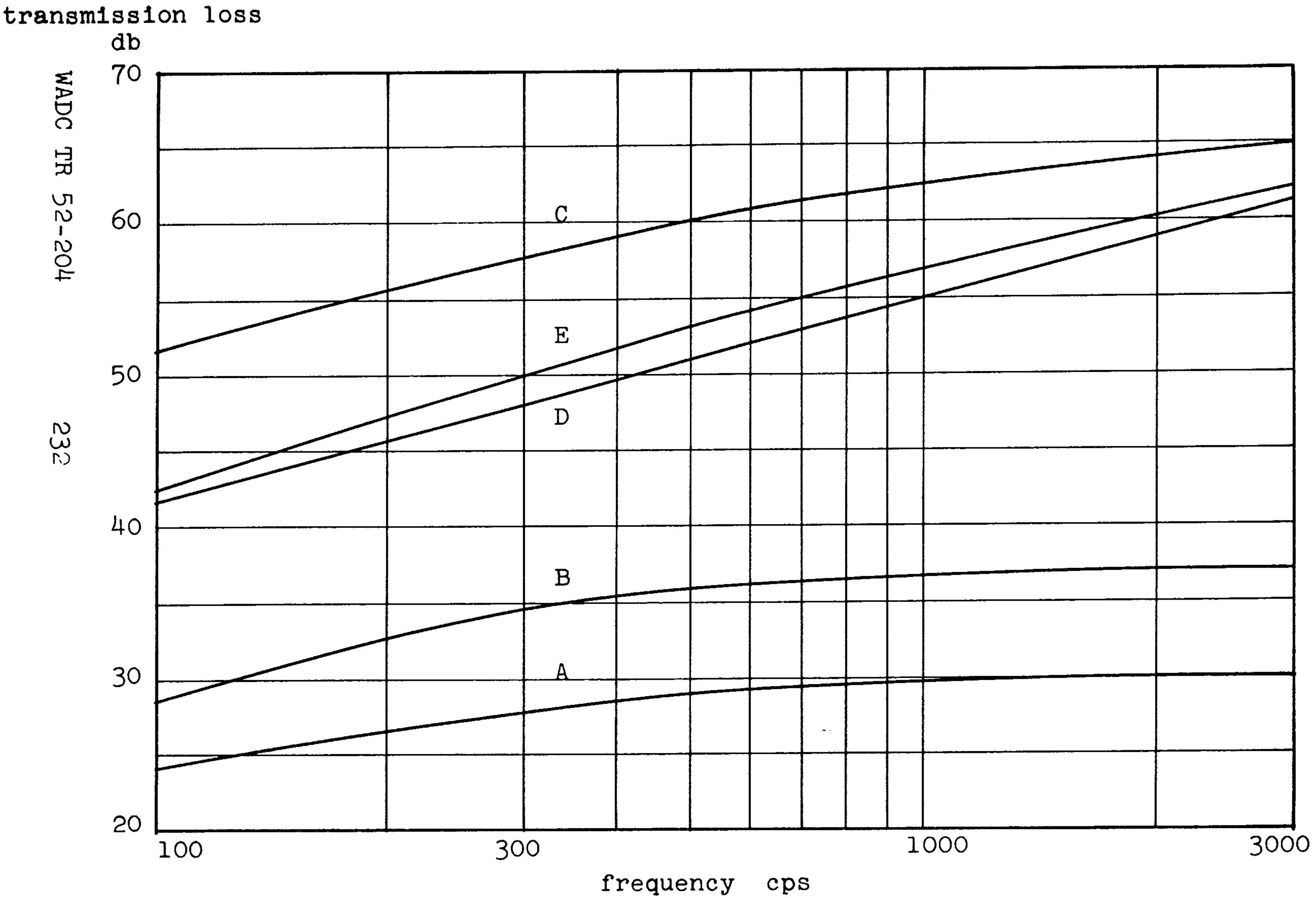


Figure 11.3

Schematic diagram of the direct and indirect sound transmission paths from a source room to a receiving room. These are

(1) direct transmission through the common wall
(2) radiation from flanking walls due to vibration induced by the motion of the common wall
(3) radiation from flanking walls due to vibration induced by sound incident on the non-common walls of the source room
(4) radiation from common wall due to vibration induced by sound incident on non-common walls of the source room
(5) indirect transmission through the flanking walls of sound radiated from the non-common walls of the source room.

values (Fig. 11.5) over the frequency range from 20 to 10,000 cps. It may also be necessary to provide floated floors, i.e., floors which are free of any solid structural bond that could act as a link for the transmission of vibration between the floated floor and its base. Such floors are often a more economical solution to the noise problem than are massive single-slab


transmission loss
db
70
60
50
40
30
20
C
E
D
B
A
100
300
1000
3000
frequency cps

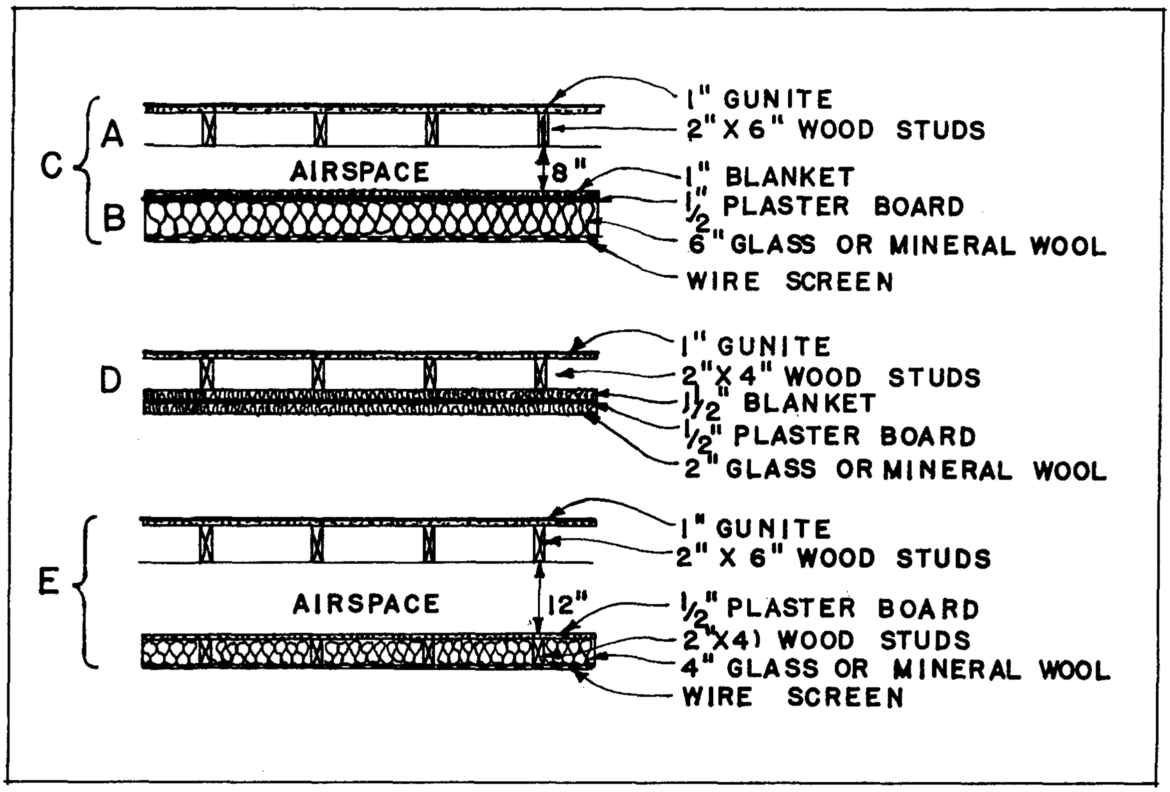


Figure 11.4

Wall constructions having large values of transmission loss.

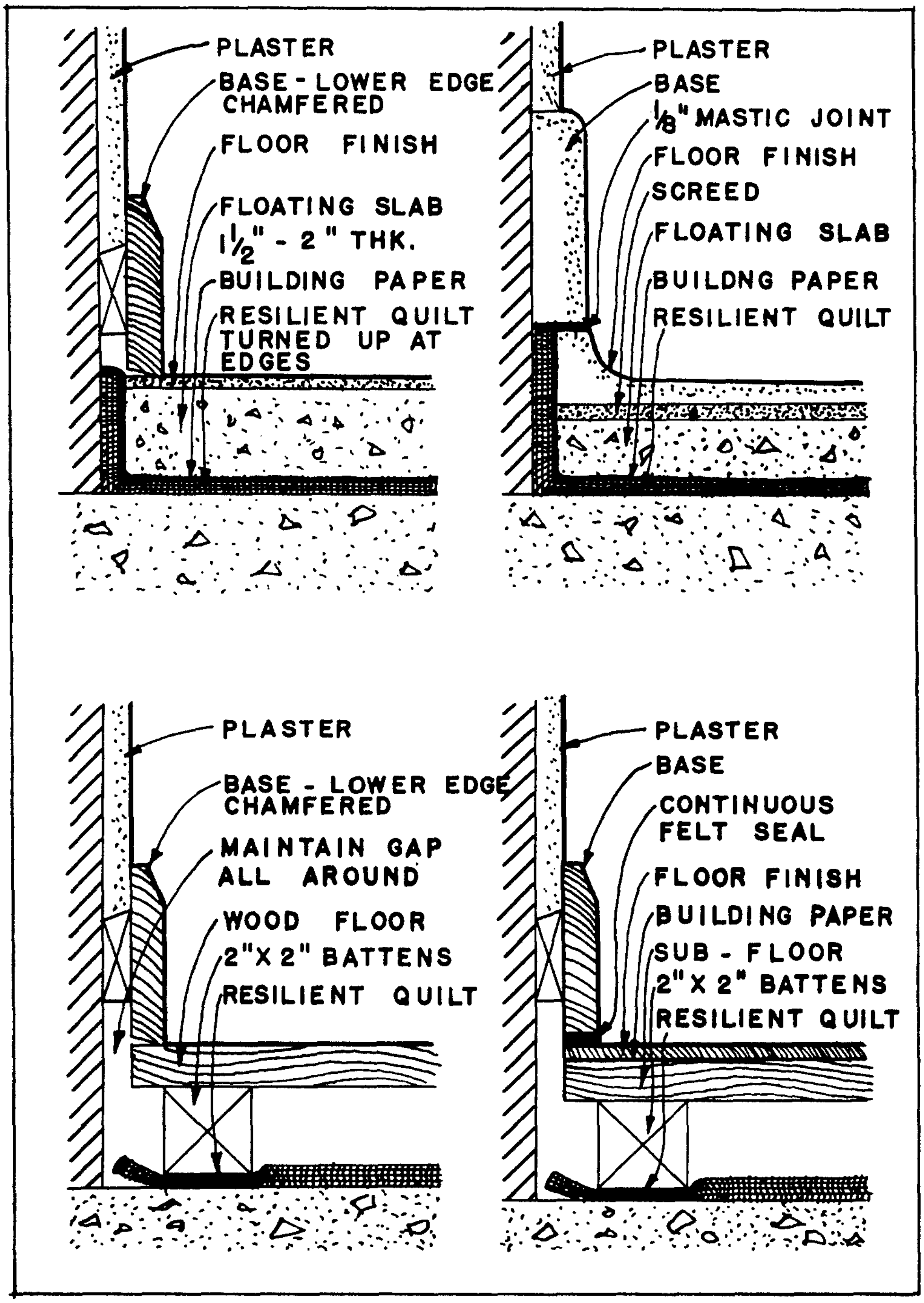
units. In the next section of this chapter several types of floating floors are discussed and their acoustical properties are reported.

11.3 Floating Floors

For a floating floor to be effective against structure-borne noise it is important that the natural resonant frequency of vibration of the floor be as low as possible. Preferably this resonant frequency should be below the lower limit of audibility. The natural frequency depends both on the weight of the floating part and on the elasticity of the resilient supporting material. The weight of the floating material is

Figure 11.5

Transmission loss for the composite walls shown in Fig. 11.4


PLASTER
BASE - LOWER EDGE CHAMFERED
FLOOR FINISH
FLOATING SLAB 1½" - 2" THK.
BUILDING PAPER
RESILIENT QUILT TURNED UP AT EDGES
PLASTER
BASE
⅛" MASTIC JOINT
FLOOR FINISH
SCREED
FLOATING SLAB
BUILDNG PAPER
RESILIENT QUILT
PLASTER
BASE - LOWER EDGE CHAMFERED
MAINTAIN GAP ALL AROUND
WOOD FLOOR
2"X 2" BATTENS
RESILIENT QUILT
PLASTER
BASE
CONTINUOUS FELT SEAL
FLOOR FINISH
BUILDING PAPER
SUB - FLOOR
2"X 2" BATTENS
RESILIENT QUILT

often determined by non-acoustical considerations so that this limits the choice of resilient materials. The best material is one which will not require a "set" under the compressional load of the floor resting on it. Also, the properties of the material should be uniform in order that the floor will receive equal support at all points without a "bridging" effect. Glass wool has been found to meet these criteria most satisfactorily.

Concrete floors, although forming an effective barrier to airborne sounds, are very poor as insulators of impact sounds so that particular attention must be paid to the design of such floors. It has been found that three main methods can be used to improve the isolation qualities of a concrete floor:

1. By the use of resilient surfacing. While this has little or no effect on the airborne sound insulation it improves the impact insulation to an extent which depends on the softness of the material used. Materials such as linoleum or asphalt mastic give very little improvement, but soft materials such as carpet on an underfelt are very useful.

2. By the use of a floating floor superimposed on the structural floor but separated from it by a layer of resilient material. This construction gives both a substantial improvement in impact sound insulation and also provides increased attenuation of airborne sounds. The floating floor is effective because vibrations caused by impact are confined to the surface which initially receives the impact and do not propagate into the surrounding structure.

3. By the use of a suspended ceiling. This serves to screen the underside of the floor receiving the impacts, thus reducing the sound reaching the room below. This improves the insulation against both airborne and impact sound by an amount depending on the surface density of the suspended ceiling and the degree of rigidity with which it is attached to the structural floor. A heavy ceiling not rigidly attached has the greatest value.

In Fig. 11.6 are shown two types of floating floors, illustrated in conjunction with three different wall skirting details. Both of these types of floating floors are designed

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Figure 11.6

Floating floors and insulated skirtings. The figure shows two floor constructions and three types of skirtings.

to be placed on top of a structural floor or a structural slab. In type (1), the layer of 1 in. thick blanket is laid directly on the base structural floor. It consists of bitumen-bonded glass wool or its equivalent with all joints of the blanket tightly butted and with the edges turned up about three inches to contain the concrete slab. The slab is 1-1/2 to 2 in. thick with a light reinforcing of open wire mesh. It is poured directly on a waterproof paper placed over the resilient blanket. Thicker blankets and thicker slabs will result in higher attenuation. The approximate transmission loss for the construction shown, i.e., 2 in. slab placed on a 4 in. structural slab of concrete, is given in Column 1 of Table 11.2.

TABLE 11.2

TRANSMISSION LOSS THROUGH COMPOSITE FLOOR STRUCTURE

Column No.	1	2
Frequency Band	2 in. concrete 1 in. glass wool blanket 4 in. concrete	3 in. concrete 4 in. glass wool blanket 6 in. concrete
20-75 cps	38 db	43 db
75-150	40	45
150-300	43	48
300-600	46	51
600-1200	51	56
1200-2400	59	64
2400-4800	67	72
4800-10000	71	76

Care must be taken to prevent perforation of the waterproof paper which will allow cement to flow into the blanket. For design purposes it should also be noted that a 2 in. slab of concrete will compress the 1 in. blanket down to approximately a 1/2 in. thickness.

In type (2), also shown in Fig. 11.6, the usual 1 in. tongue and groove or plane-edge boarding of the sub-floor is nailed to battens which measure at least 1-1/2 in. by 2 in. These battens rest on at least two layers of the resilient supporting material. This material may be in the form of

strips under the battens or may be a blanket placed over the entire floor. Finished floors laid over this construction should be laid over building paper placed on the sub-floor for an air seal. Care must be taken to ensure that no nails in the flooring or in the battens themselves protrude into or through the resilient blanket. Nailing the battens to the structural slab would seriously reduce the insulation value of the floor. Where the structural slab is 4 in. concrete, this construction will yield transmission loss values in each frequency band which are 6 db less than those tabulated in Table 11.2.

In both types (1) and (2) of Fig. 11.6 special attention should be paid to such details as piping which passes through the slab or heating or electrical piping placed under the floor. Electrical, gas or water conduits should be placed beneath the resilient blanket. Heating pipes, however, should be placed on top of the paper covering before the slab is poured. All pipe connections to the main structure should be made with non-rigid fittings. This will permit the natural expansion and contraction of the slab and also the pipes will not act as carriers of structure-borne noise. Pipes should be wrapped with a bituminous felt along the length which passes through the slab to prevent any free air paths. Wherever possible, all partition walls should be carried through to the structural floor.

A third type of floating floor construction, shown in Fig. 11.7, is used where large noise reduction values are required. In this case there is a 3 in. layer of concrete laid on four layers of a resilient blanket material whose nominal thickness is 1 in. and which are in turn placed on a structural slab of 6 in. concrete. The finish floor is of asphalt tile laid on a 1/4 in. layer of mastic. The same attention must be paid to edge details as was mentioned for types (1) and (2) above. This type of floor is almost a requirement for such areas as the floors of control room located adjacent to aircraft engine test cells. The approximate transmission loss values to be expected from this type of floor are tabulated in column 2 of Table 11.2.

11.4 Suspended Ceilings

Suspended ceilings placed below structural floors can be used to improve the isolation qualities of a concrete floor. Suspended ceilings do not prevent impact noises from passing down the walls of a room from the floor above unless an

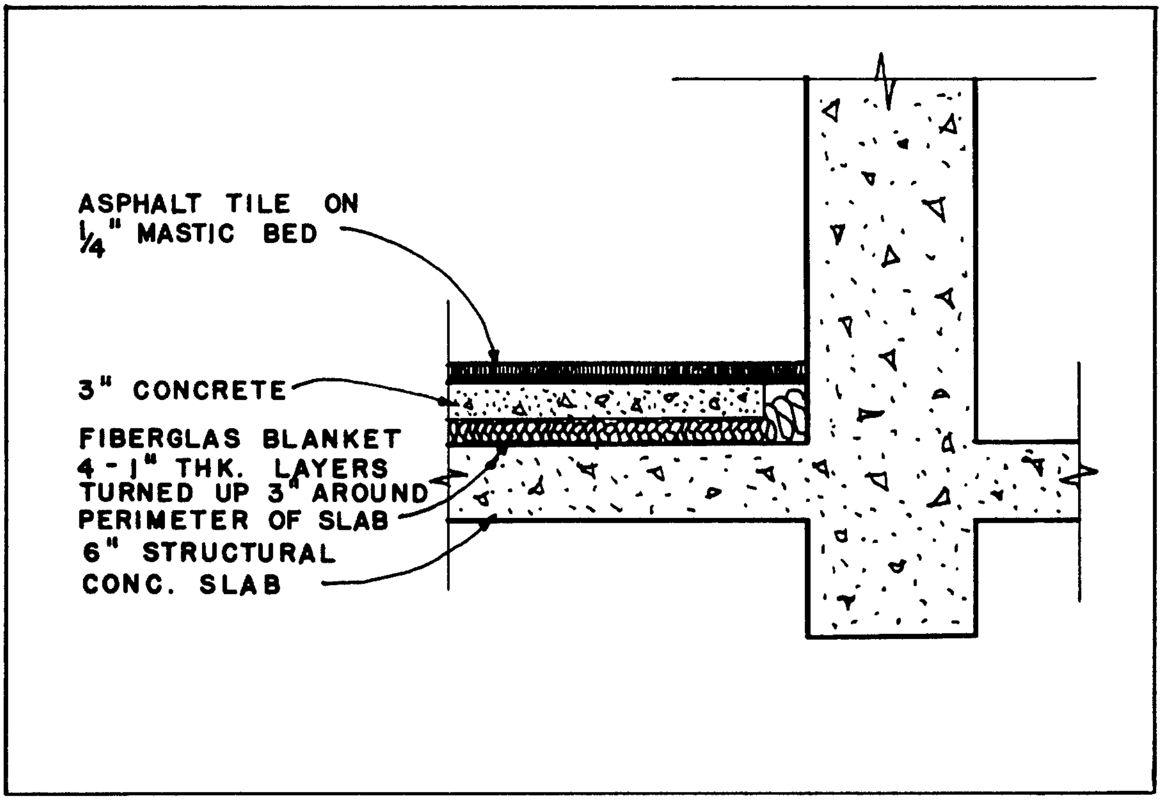


Figure 11.7

A floating floor construction used when large noise reduction values are required.

isolating mounting is provided at the junction of the wall and the structural slab. If a resilient pad is not provided at this point, suspended ceilings cannot alone provide the same degree of isolation as a floating floor. They can, however, make a useful addition to the insulation against direct transmission through the floor. Moreover, suspended ceilings also provide a definite increase in insulation against airborne sound.

Suspended ceilings may be constructed on battens held in clips close to the structural floor or may be suspended from hangers as shown in the two detailed sections of Fig. 11.8. The former are more correctly called "furred" ceilings and the separation between ceiling and slab usually does not exceed 1 in. The latter "hung" ceiling usually provides a separation of the order of 12 in. The relative sound insulation afforded by these constructions depends largely on three factors: the

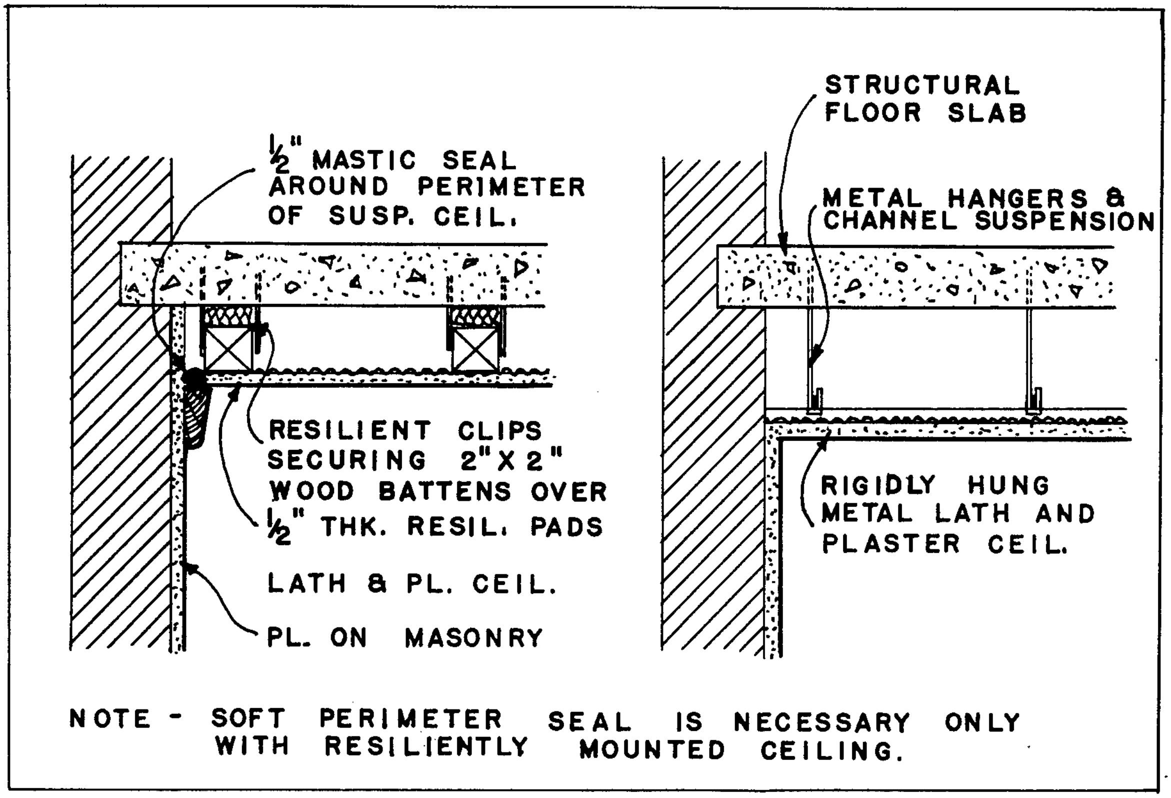


Figure 11.8

Two types of suspended ceiling construction.

depth of the air space between ceiling and slab, the surface density of the ceiling proper, and the type of suspension or mounting system used. In general, a "heavy" ceiling (5 lb/sq ft or more), a separation of about 12 in. or more and resilient suspension will give the best results.

Resilient suspension may be accomplished in several ways. Continuous felt or cork pads may be inserted under wood or metal battens ("furring") as shown in Fig. 11.8. Patented clips incorporating a spring action may be used in conjunction with the metal channel grid supporting the ceiling. Or finally, patented rubber-in-shear vibration isolating hangers may be employed. It should be noted that if a resilient suspension system is used the boundaries of the suspended ceiling must not be bonded or otherwise regidly attached to the surrounding walls. A small (1/2 in.) separation must be maintained through-

out at this point and may be sealed with a soft mastic. Similarly, ductwork, electrical conduit, piping, etc., penetrating the ceiling should not be rigidly attached to the slab above or the resiliency of the ceiling construction will be largely nullified. Penetrations of the ceiling by lighting elements should be minimized and should in any case be back-plastered or otherwise enclosed to prevent the leakage of sound through the ceiling.

Concrete ribbed floors which have a recessed soffit of the type shown in Fig. 11.9 usually involve a separate ceiling. Here the small size of the area of contact between the structural floor and the ceiling immediately aids in achieving some noise reduction. If the ceilings are of the order of 5 lb/sq ft and resilient pads are inserted at points of contact between the concrete slab and the ceiling construction the transmission loss to be expected will be as given in Table 11.3.

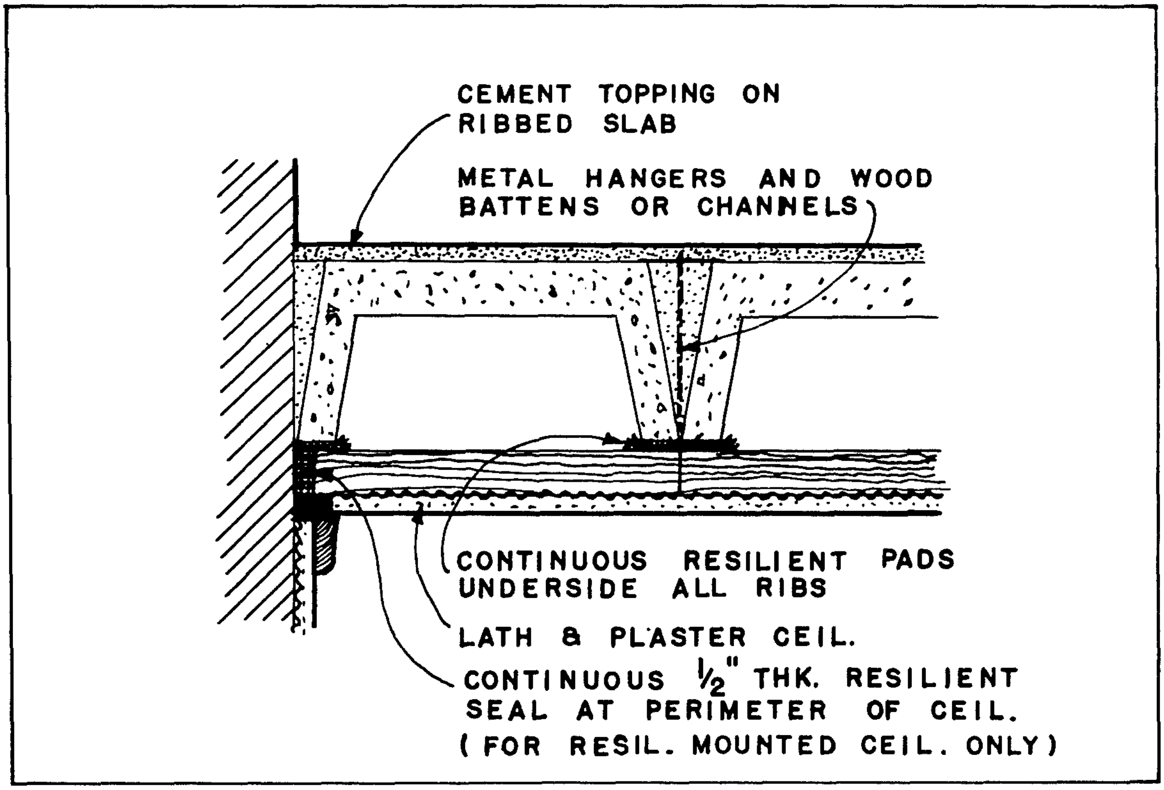


Figure 11.9

A hollow soffit structural floor.

Suspended ceilings and floating floor combinations employing steel joists have not been explored extensively, but a suggested method is shown in Fig. 11.10. The steel joists should be isolated from both the floor and ceiling by resilient blankets. The floating floor must be placed so that it does not come into contact with either the joists or the sidewalls. Also the metal lath and plaster ceiling should be reinforced by a ribbed metal mesh. Transmission loss values of the same order of magnitude as those given in Table 11.4 may be expected if the floor slab is of the order of 3 in. thick and the ceiling is about 8 lb/sq ft.

TABLE 11.3

TRANSMISSION LOSS THROUGH CONCRETE FLOOR AND SUSPENDED CEILING

(4 INCH CONCRETE AND A 5 LB/SQ FT CEILING)

Frequency Band	TL
20-75 cps	68 db
75-150	70
150-300	73
300-600	76
600-1200	Above 80
1200-2400	Above 80
2400-4800	Above 80
4800-10000	Above 80

11.5 Transmission of Sound Through Cylindrical Pipes

At sufficiently low frequencies the pressure distribution inside a cylindrical pipe is uniform regardless of the source of excitation. In this case we can say that the walls move in Hooke's Law type motion under the influence of an alternating hydrostatic pressure. The effectiveness of the wall in reducing the SPL can be obtained by measuring the SPL at the inner and outer surfaces of the wall. In practice this type of measurement is difficult and some indirect means must be employed. However, we can make a theoretical estimate of how much noise reduction is to be expected. For pure tones the NR is related to the transmission loss of the cylindrical pipe.

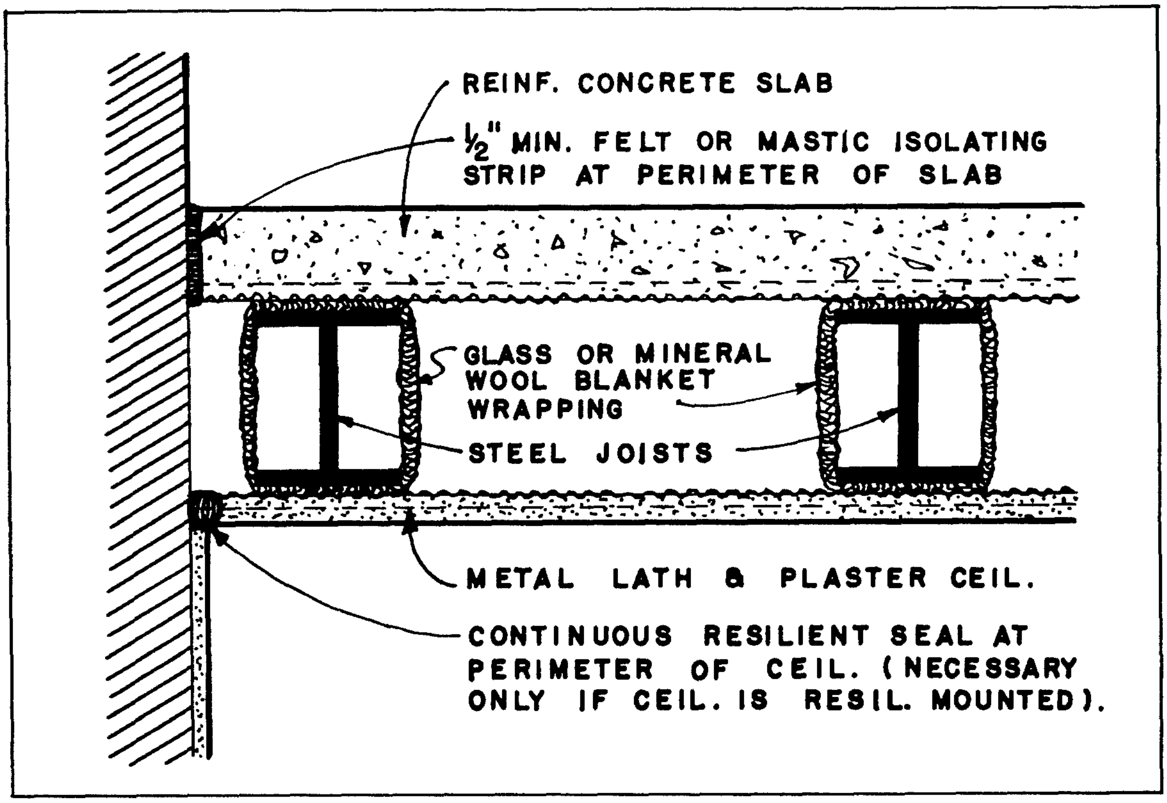


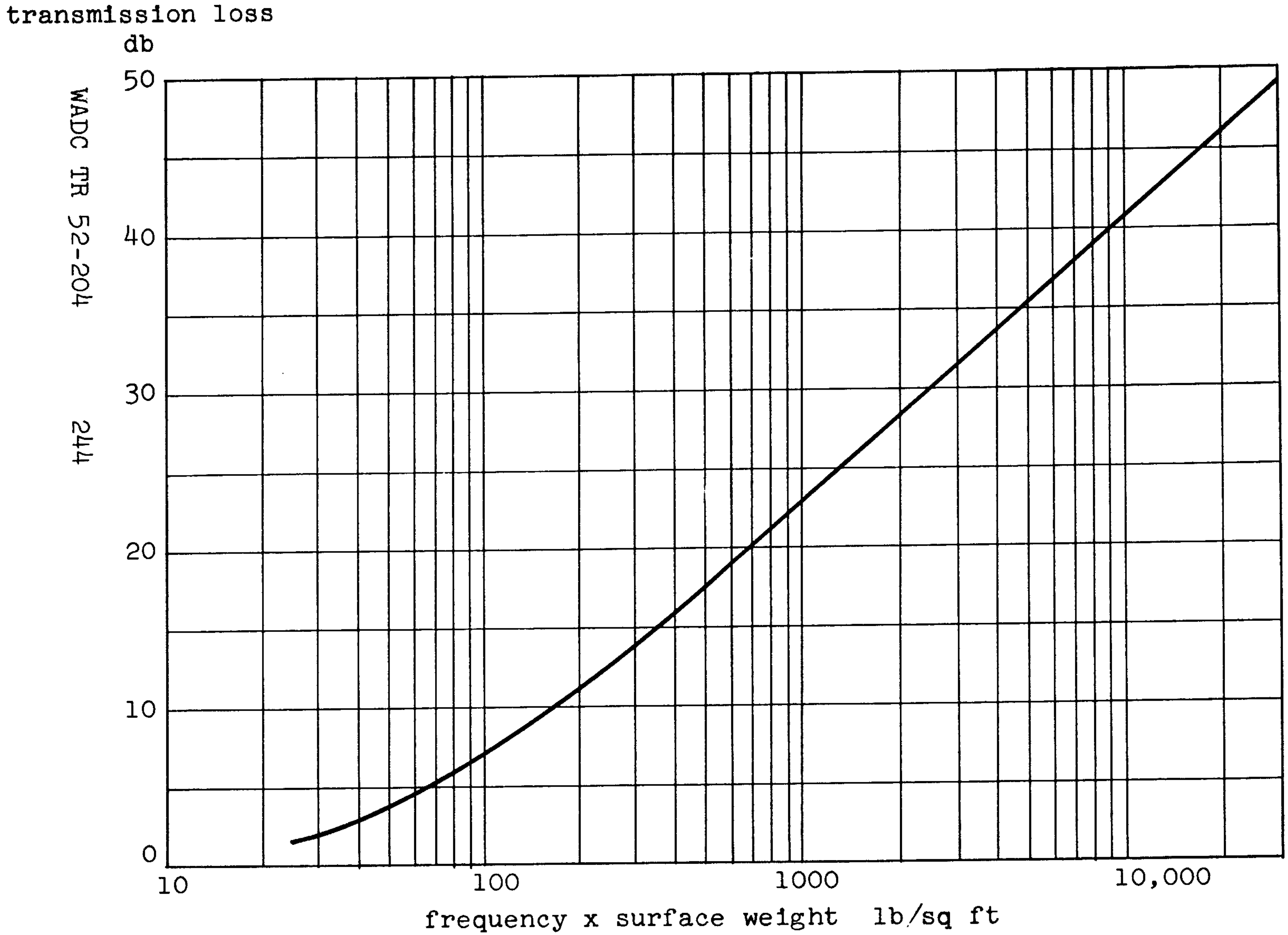
Figure 11.10

Method of isolating the floor and ceiling below.

It can be shown that the expected NR is given by

$$NR = 10 \log_{10} [1 + (k_o t_1)^2 (\gamma - \frac{1}{\gamma})^2] \quad \text{db}$$

in which $k_o = f_o/c$ is the ratio of the natural frequency of the wall to the velocity of sound in the medium inside and outside the cylinder, $t_1 = \sigma t_s/\rho$, the product of the density of the wall material and the wall thickness divided by the medium density; and $\gamma = \omega/\omega_o$ is the ratio of the angular driving frequency to the natural frequency of the wall. The natural frequency of the wall for this motion can be given in the simple form $f_o = \omega_o/2\pi = 5400/d$ where d is the inside diameter of the pipe, in feet.


transmission loss
db
50
40
30
20
10
0
10
100
1000
10,000
frequency x surface weight lb/sq ft

Several laboratory and field measurements have been checked against the Eq. (11.13). It is found in all cases that the resonance is properly located; however, the predicted NR is higher than the measured NR by as much as 50% in db. The reason for this can be traced to asymmetries in the walls of the cylinder. The assumption of a perfectly symmetrical and seamless pipe is seldom met in practice. As a result of this and other physical considerations the pipe will not stretch as supposed, since the tendency to bend is very great. The bending motion cannot be specified and hence this phase of the description is indeterminate.

If one assumes that the walls will bend it is possible to advance a plausibility argument and thus obtain a NR due to bending. If air is pumped into a cylinder creating an excess pressure Δ p, the cylinder maintains its shape. Now let this pressure be reduced to zero, the shape being unchanged, and further reduced until a negative excess pressure is applied. It is here that the cylinder takes on an elliptical shape. The excess pressure goes positive again completing a cycle while the cylinder has undergone only one-half cycle in its motion. One can see then that for a driving frequency twice the bending frequency of the wall, an "in phase condition" occurs at which there should be complete transmission. The lowest possible bending frequency of the cylinder is $f_1 = 800\ t_2/d_1^2$, in which t_2 is the thickness of the wall in inches and d_1 the diameter of the pipe in feet. We take as the bending frequency twice this value or $f_B = 1600\ t_2/d_1^2$. This value is substituted into the previous relation to give

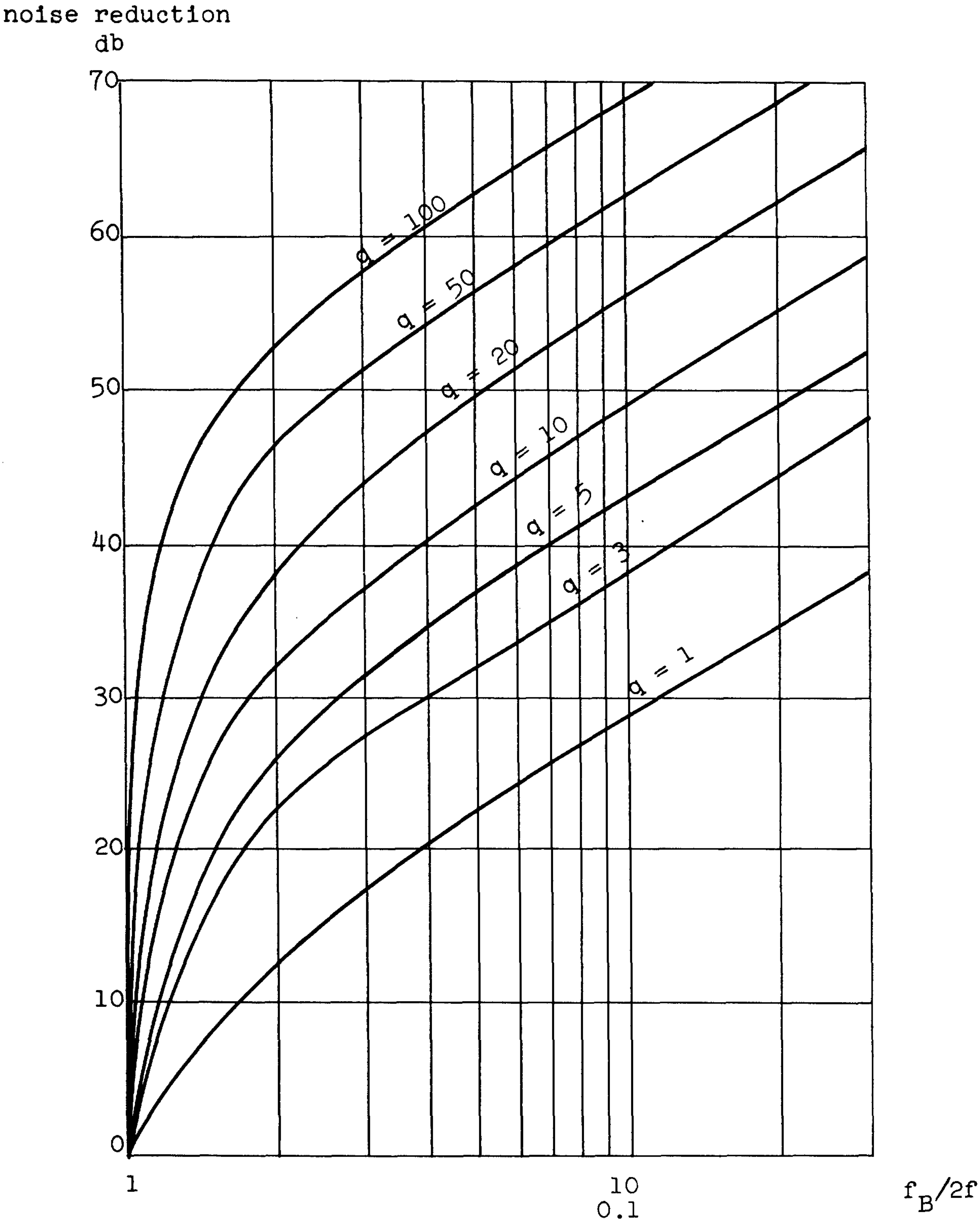
$$(NR)_B = 10 \log_{10} \left[1 + (k_B\ t_1)^2 \left(\gamma_B - \frac{1}{\gamma_B}\right)^2\right] \text{db} \qquad (11.14)$$

as the NR for bending.

The formula above has also been checked against laboratory and field measurements and it is found that this may be low by as much as 15% (in db), in the range $f < f_B$ and that f_B is not generally measured. The principal value of this calculation has been then to reduce the NR at low frequencies to a more reasonable value compared with measured values.

Figure 11.11

Theoretical "Mass Law" intensity reduction for cylinders.


noise reduction
db
70
60
50
40
30
20
10
0
q = 100
q = 50
q = 20
q = 10
q = 5
q = 3
q = 1
1
10
0.1
$f_B/2f$

A procedure can be suggested which though empirical will give an estimate of the NR. Below $f_B/2$ the NR curve decreases with frequency at 6 db/octave due to stiffness. If a line having this same slope is extended to $f = 5400/d$, since this resonance seems to be present, the resulting NR will be everywhere less than the measured NR.

At about an octave above the stretching frequency, Eq. (11.13) shows that the NR should increase at 6 db/octave due to the mass of the cylinder. There are charts available which give the TL of solid damped partitions as a function of their mass per unit area (Fig. 11.11) for both reverberant and normally incident sound. Experiment shows that if the reverberant values are reduced by 8 db, agreement is obtained. There seems to be no physical basis for this procedure. The reverberant mass law increases at 5 db/octave instead of 6 db/octave which is the slope for the normal incidence mass law. Again as an engineering procedure the mass law is extrapolated back to the stretching frequency and joined to the bending curve. This then is the estimate of the NR for pure tones over the entire range of frequencies.

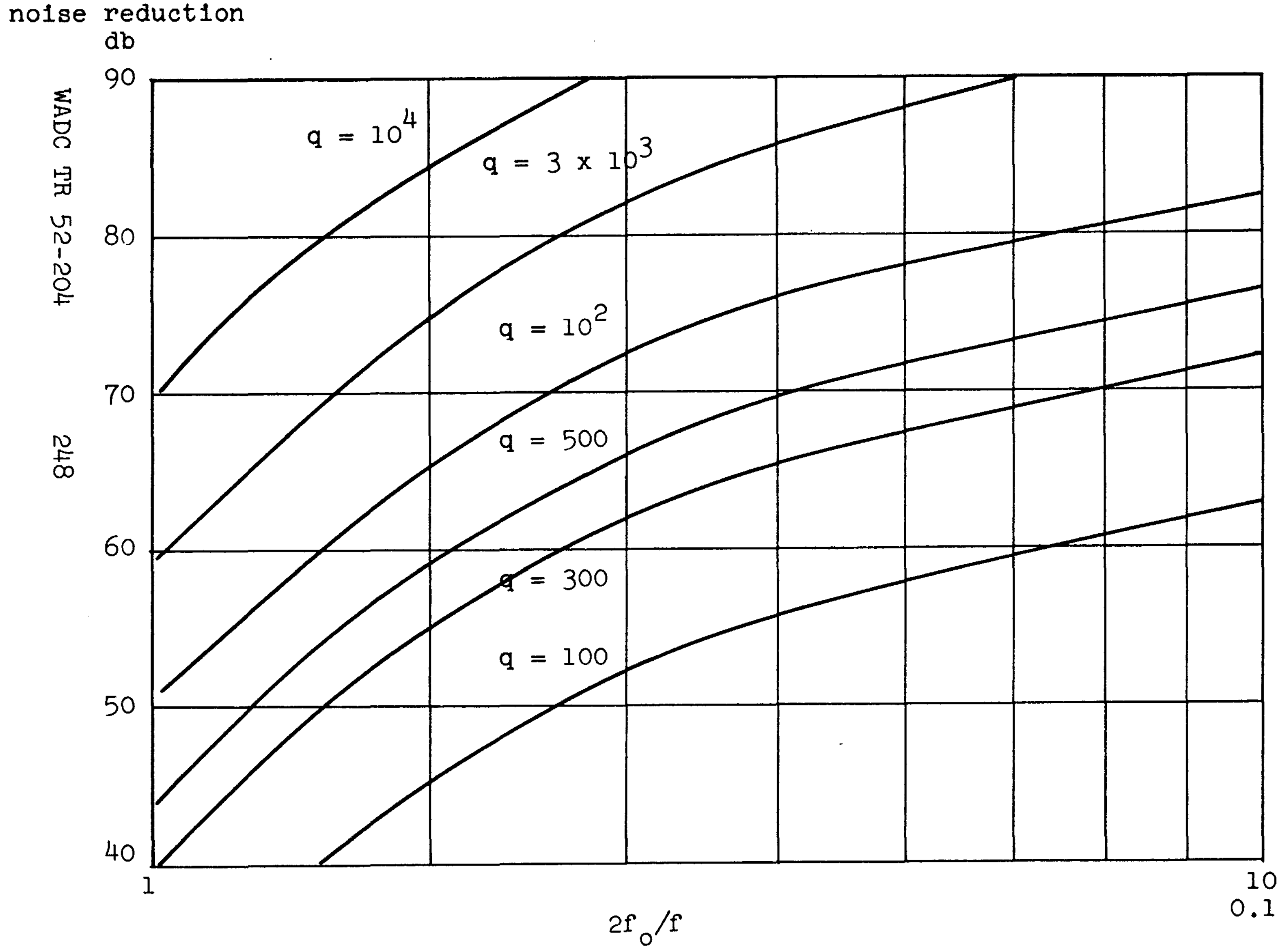
Design charts, Figs. 11.12 and 11.13, have been included in order to facilitate calculations according to the procedure above. These charts give the NR in db at $f = f_B/2$ and $f = 2f_o$. The first value is obtained by entering the chart at $\gamma = f_B/2f$ and reading the NR for $q = f_B t_2$. The second value is obtained by using $\gamma = 2f_o/f$ for $q = f_o t_2$. Interpolation is valid at these two points and is accomplished by the formula:

$$\frac{(NR)_x}{NR} = 20 \log_{10} (q_x / q),$$

in which the quantity NR is desired for the particular value of q, and $(NR)_x$ is given in the charts.

Figure 11.12

Noise reduction through cylinders as a function of $f_B/2f$ for various values of q.


noise reduction
db
90
80
70
60
50
40
1
10
0.1
2f o/f
q = 10^4
q = 3 x 10^3
q = 10^2
q = 500
q = 300
q = 100

11.6 Noise Reduction by Wrappings

In order to reduce sound transmission through tube walls, wrappings are often used. Normally this wrapping is composed of a porous layer covered with an impervious skin, and is applied directly to the tube wall. The porous layer (PF Fiber-board, Aerocor or some similar material) usually has a thickness of 1 to 6 inches and weighs between 4 and 10 lb/cu ft. The skin is usually sheet metal weighing about 1 lb/sq ft.

In the following analysis of the noise reduction, the wrapping has been considered as a plane wall. This assumption is good for rectangular tubes and also for cylindrical tubes with diameters much larger than the thickness of the wrapping. The general analysis of the cylindrical wrapping is somewhat involved and should give values only slightly different from the plane case.

Let the porous layer be described by the following parameters:

Mass per unit volume $= \rho_1$

Mass per unit area $= m_1 = \rho_1 t_1$

Porosity $= P$

Thickness $= t_1$

Flow resistance $= r$

The skin parameters are:

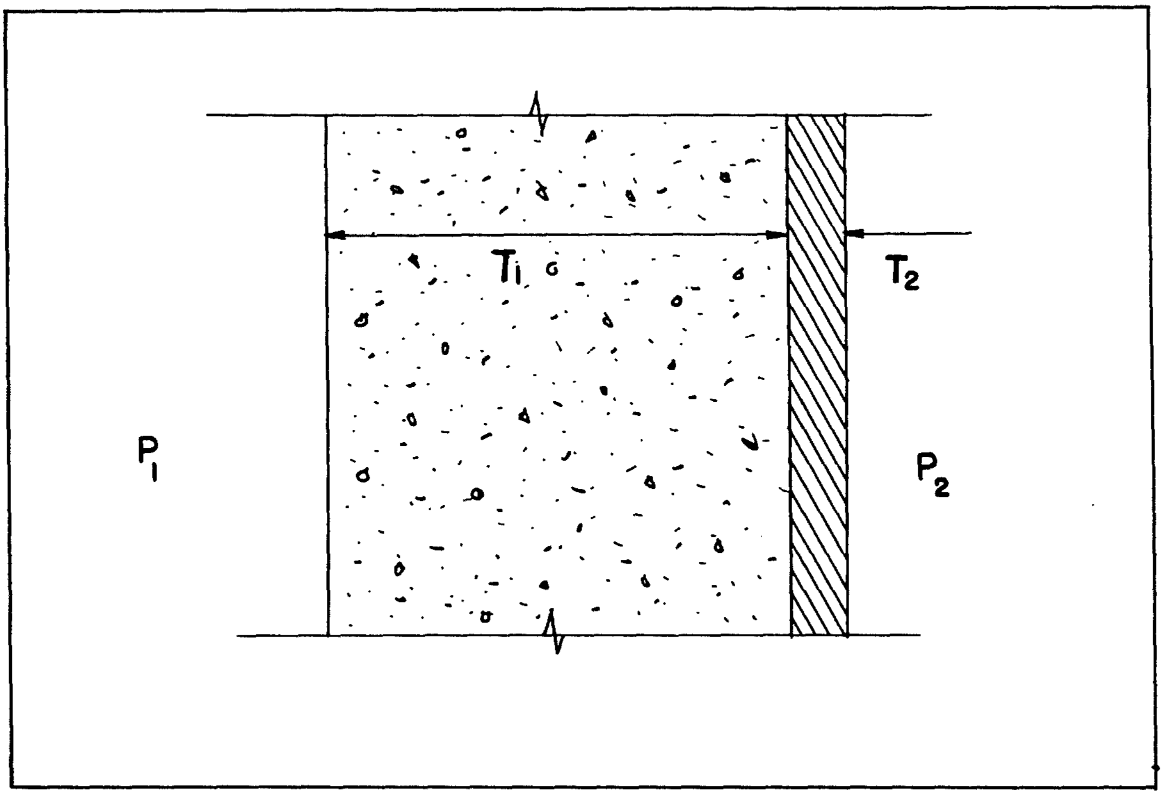
Mass per unit volume $= \rho_2$

Mass per unit area $= m_2 = \rho_2 t_2$

Thickness $= t_2$.

Figure 11.13

Noise reduction through cylinders as a function of $2f_o/f$ for various values of q


T1
T2
P1
P2

$$\theta = \frac{rt_1}{\rho_o c}$$

$$\beta_1 = \frac{\rho_1}{\rho_o}$$

$$\beta_2 = \frac{\rho_2}{\rho_o}$$

ρ_o = density of air

($\rho_o c$ = characteristic impedance of air).

The propagation of sound in a porous layer is rather complex, since a complete treatment must consider both an air-borne wave and a structure-borne wave 9,10/. However in most cases, and in particular at high frequencies, the simplifying assumption can be made that the porous structure is rigid. At very low frequencies however, it is necessary to consider the motion of the structure itself in addition to the air motion in the material.

In this analysis we have thus taken into account motion of the structure only at low frequencies but assumed a rigid structure at high frequencies.

First let the porous structure be considered rigid. In that case the ratio between the sound pressures on the front side and the back side of the wrapping can be shown to be:

Figure 11.14

Porous pipe wrapping with impervious outer cover

$$\frac{p_1}{p_2} = \frac{Z}{\rho_o c} \frac{\cosh\,(-ikt_1 + \psi)}{\sinh \psi}$$

in which:

$$\tanh \psi = \frac{Z}{\rho_o c} \frac{1}{1 - i\beta_o k_o t_2} ,$$

$$k_o = \omega/c \qquad k = \frac{\omega}{c}\sqrt{1 + i\,\frac{\theta}{k_o t_1}} ,$$

$$Z = \rho_o c \sqrt{1 + i\,\frac{\theta}{k_o t_1}} .$$

The noise reduction coefficient is then obtained as

$$NR = 20 \log \left[\frac{p_1}{p_2}\right] .$$

Over most of the frequency range $\beta_2 k_o t_2 \gg 1$,

and hence:

$$\psi \simeq i\,\frac{1}{\beta_2 k\, t_2} ,$$

and

$$NR = 20 \log \left[\frac{Z}{\rho c}\right] + 20 \log \beta_2 k t_2 + 20 \log \left[\cosh\,(-ikt_1)\right], \quad (11.15)$$

in which the second term, the transmission loss for the skin, is usually predominant. For frequencies sufficiently high, $k_0 t_1 \gg \theta$, and Eq. (11.15) reduces to:

$$NR \simeq 20 \log (\rho_2 k t_2) + 10 \log \quad (\cosh \theta + \cos 2k_0 t) - 3 \text{ db.}$$

Low Frequency Approximation. At low frequencies, $k_0 t_1 \ll 1$, the motion of the porous layer itself has been included and the equivalent circuit for the wrapping is that shown in Fig. 11.15.

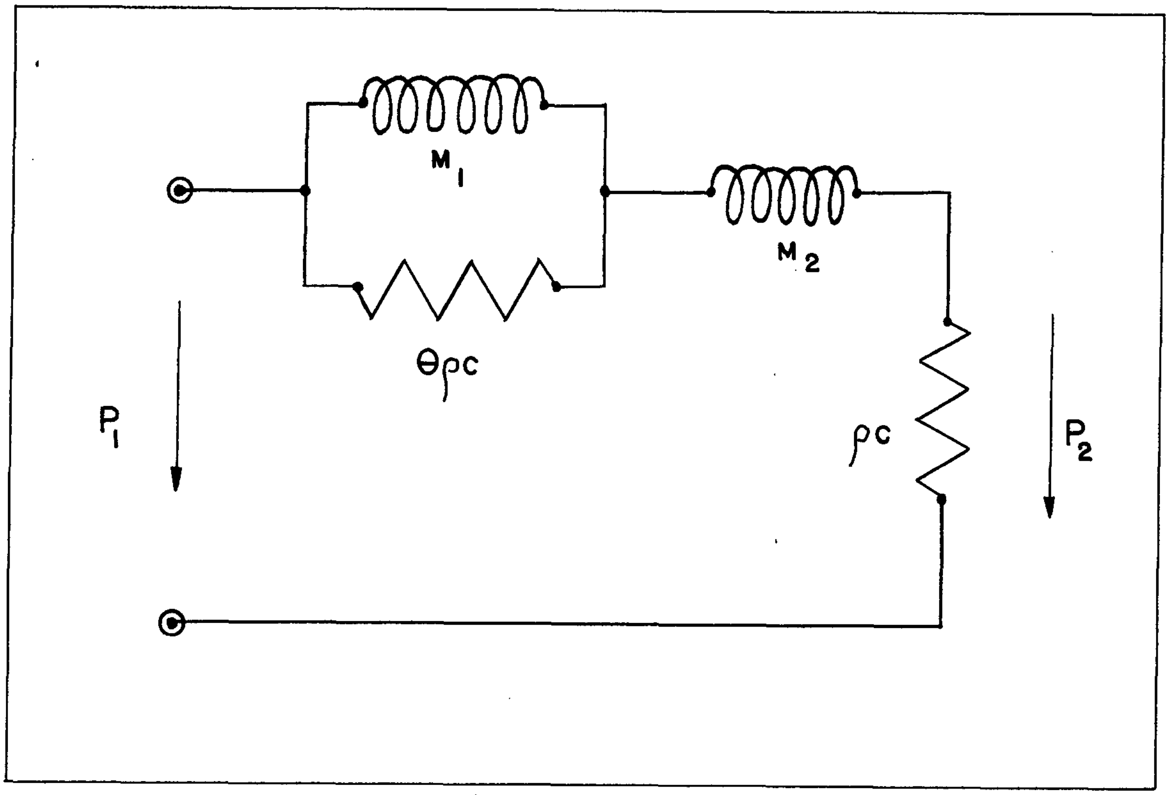


Figure 11.15

Equivalent circuit for wrapping and outer cover which is valid at low frequencies

The noise reduction can now be written:

$$NR = 20 \log \beta_2 k_o t_2$$

$$+ 10 \log \left[\frac{\left(\frac{\theta}{\beta_2 k_o t_2} \quad \frac{1}{\beta_1 k_o t_1} - 1\right)^2 + \left(\frac{\theta}{\beta_1 k_o t_1} + \frac{\theta + 1}{\beta_2 k_o t_2}\right)^2}{1 + \left(\frac{\theta}{\beta_1 k_o t_1}\right)^2} \right]$$

in which the first term usually is predominant. In fact, when $\beta_2 k_o t_2 \gg \theta + 1$ and $\beta_1 k_o t_1 \gg \theta$, the NR reduces to:

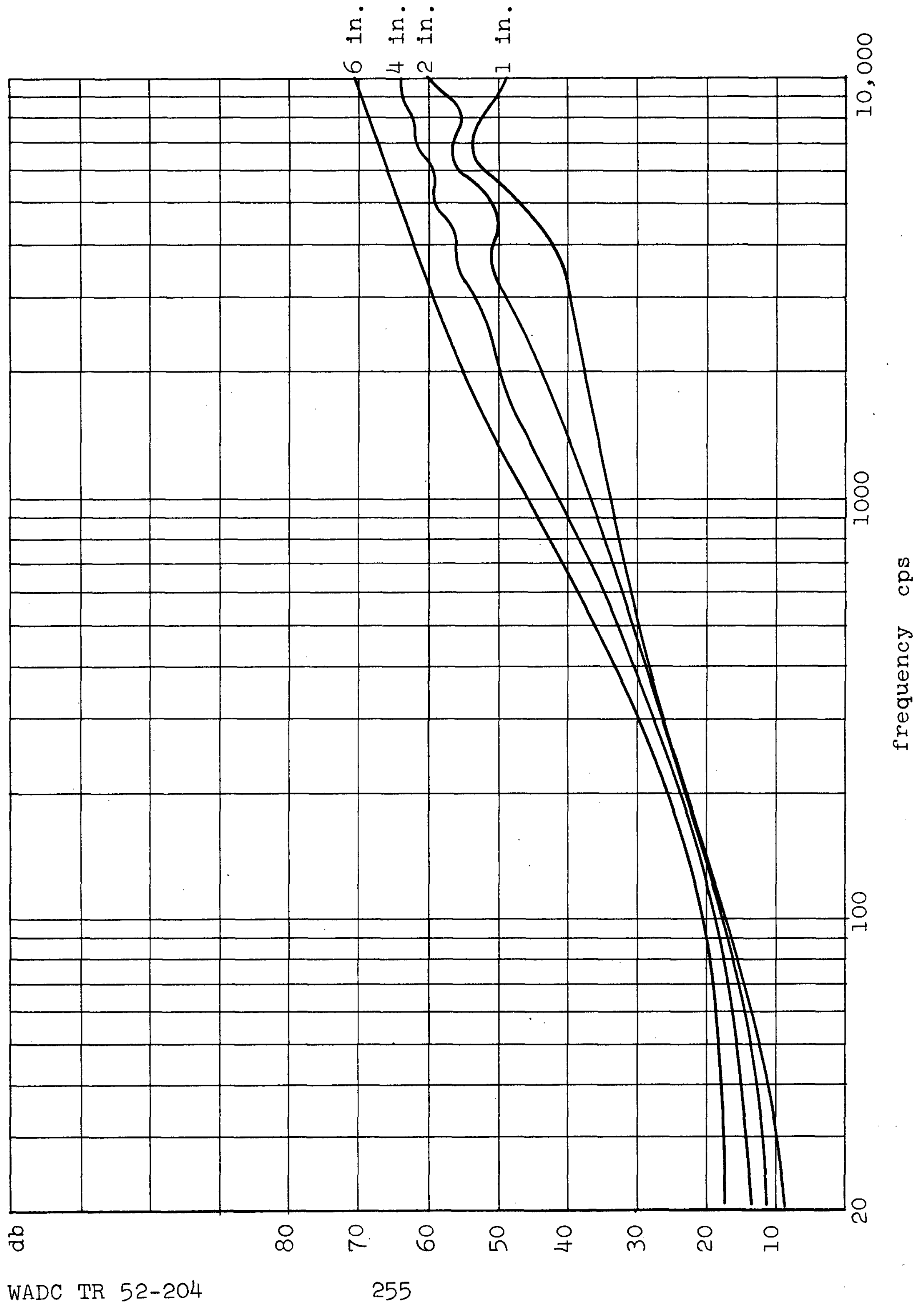
$$NR = 20 \log \beta_2 k_o t_2 + 4.3 \left(\frac{\theta}{\beta_1 k_o t_1} + \frac{\theta + 1}{\beta_2 k_o t_2}\right) \quad \text{db.}$$

Numerical Result. The noise reduction has been plotted as a function of frequency for a few cases of practical interest, shown in Fig. 11.16 and 11.17. In Fig. 11.16 the porous material consists of Fiberboard, 6 lbs/ft^3 and thicknesses from 1 in. to 6 in.; Fig. 11.17 refers to a density of 10.5 lbs/ft^3 and thicknesses between 1 in. and 4 in. The effect of the porous layer is particularly important at very low and very high frequencies, whereas in the frequency range between 100 and 300 cps the noise reduction is almost independent of thickness.

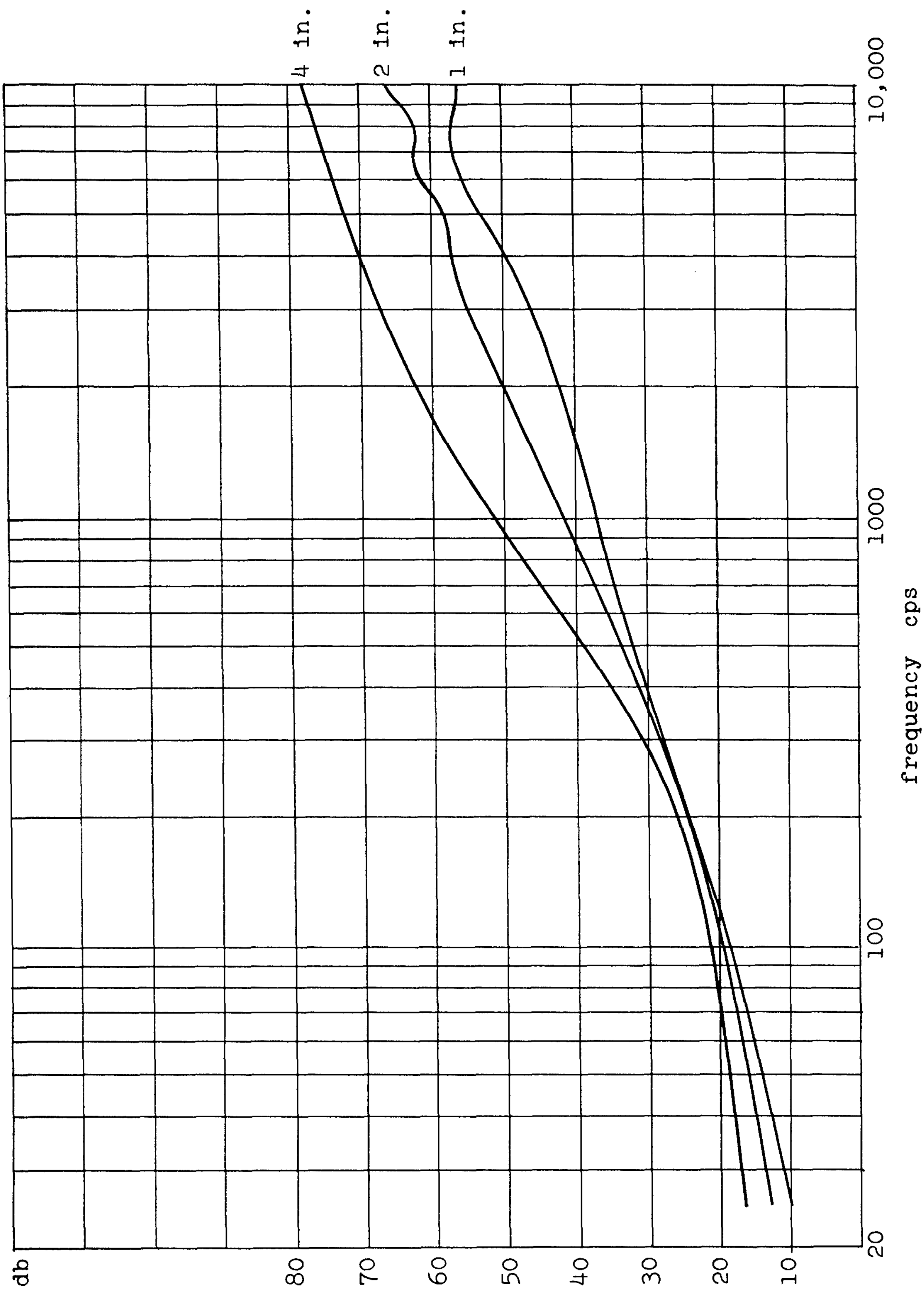
Noise Reduction by Porous Layer Alone. The noise reduction for the porous blanket alone is:

Figure 11.16

Noise reduction of various thicknesses of 6 lb/ft^3 PF Fiberglas covered with an impervious skin weighing 1 lb/ft^2



6 in.
4 in.
2 in.
1 in.
10,000
1000
100
20
frequency cps
db
80
70
60
50
40
30
20
10



4 in.
2 in.
1 in.
db
80
70
60
50
40
30
20
10
20
100
1000
10,000
frequency cps

$$NR = 20 \log \left[\frac{p_1}{p_2} \right] = 20 \log \quad \cosh\left(-ikt_1\right) + \frac{Z}{\rho_o c} \sinh\left(-ik_o t_1\right)$$

If $k_o t_1 \gg \theta$ this expression becomes:

$$NR = 4.3\ \theta \ \ db\ .$$

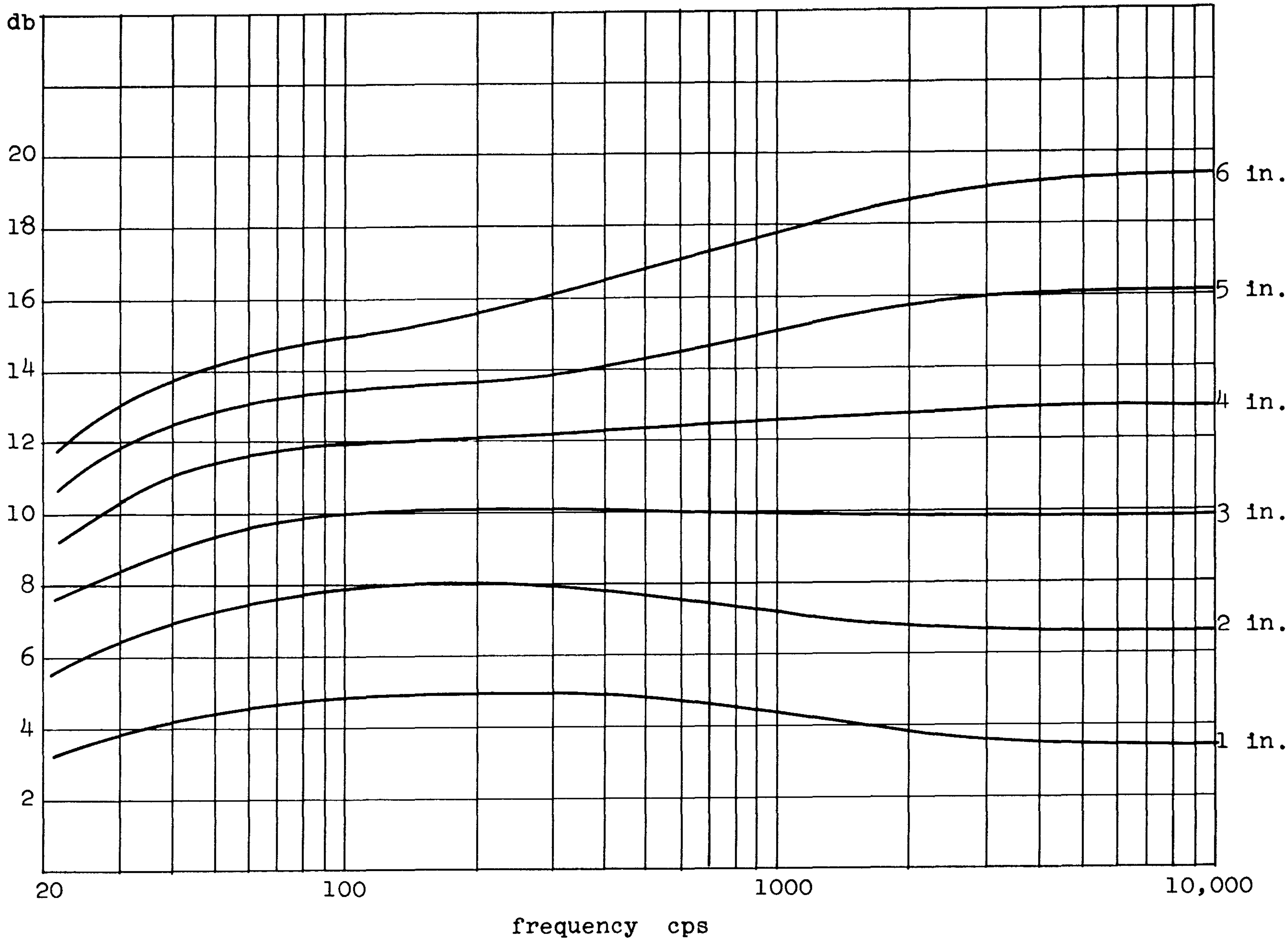
At low frequencies, $k_o t_1 \ll 1/\theta$ and $k_o t < \theta$, the equivalent circuit for the blanket is as shown in Fig. 11.18, and the noise reduction becomes:

$$NR = 10 \log \left[\frac{1 + \left(\frac{\beta_1 k t_1}{\theta}\right)^2 \quad (1 + \theta)^2}{1 + \left(\frac{\beta_1 k t_1}{\theta}\right)^2} \right] .$$

In Fig. 11.19 and 11.20 the noise reduction for layers of PF Fiberboard has been plotted as a function of frequency for several different thicknesses varying from 1 in. to 6 in. and for two different densities 6 and 10.5 lbs/ft^3. It is evident that the porous layers alone offer a comparatively small noise reduction, and are not of much help without an impervious membrane as a cover.

Figure 11.17

Noise reduction of various thicknesses of 10.5 lb/ft^3 PF Fiberglas covered with an impervious skin weighing 1 lb/ft^2


db
20
18
16
14
12
10
8
6
4
2
20
100
1000
10,000
frequency cps
6 in.
5 in.
4 in.
3 in.
2 in.
1 in.

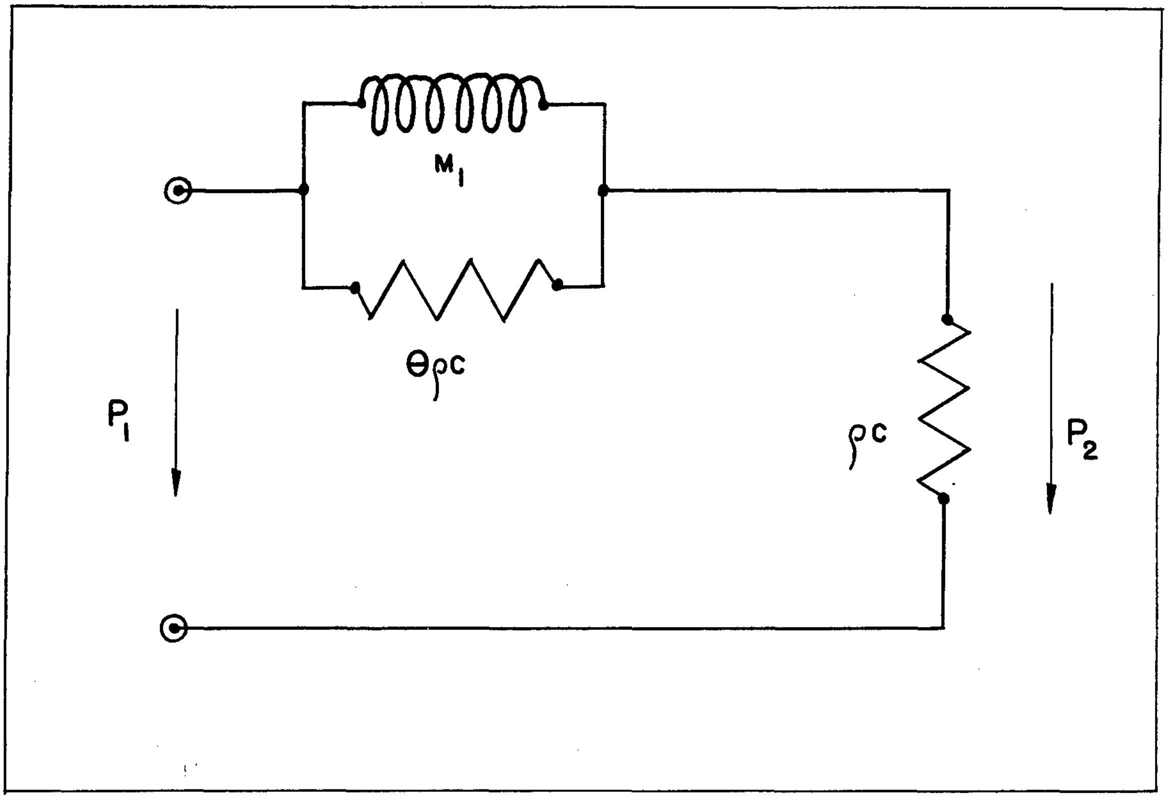
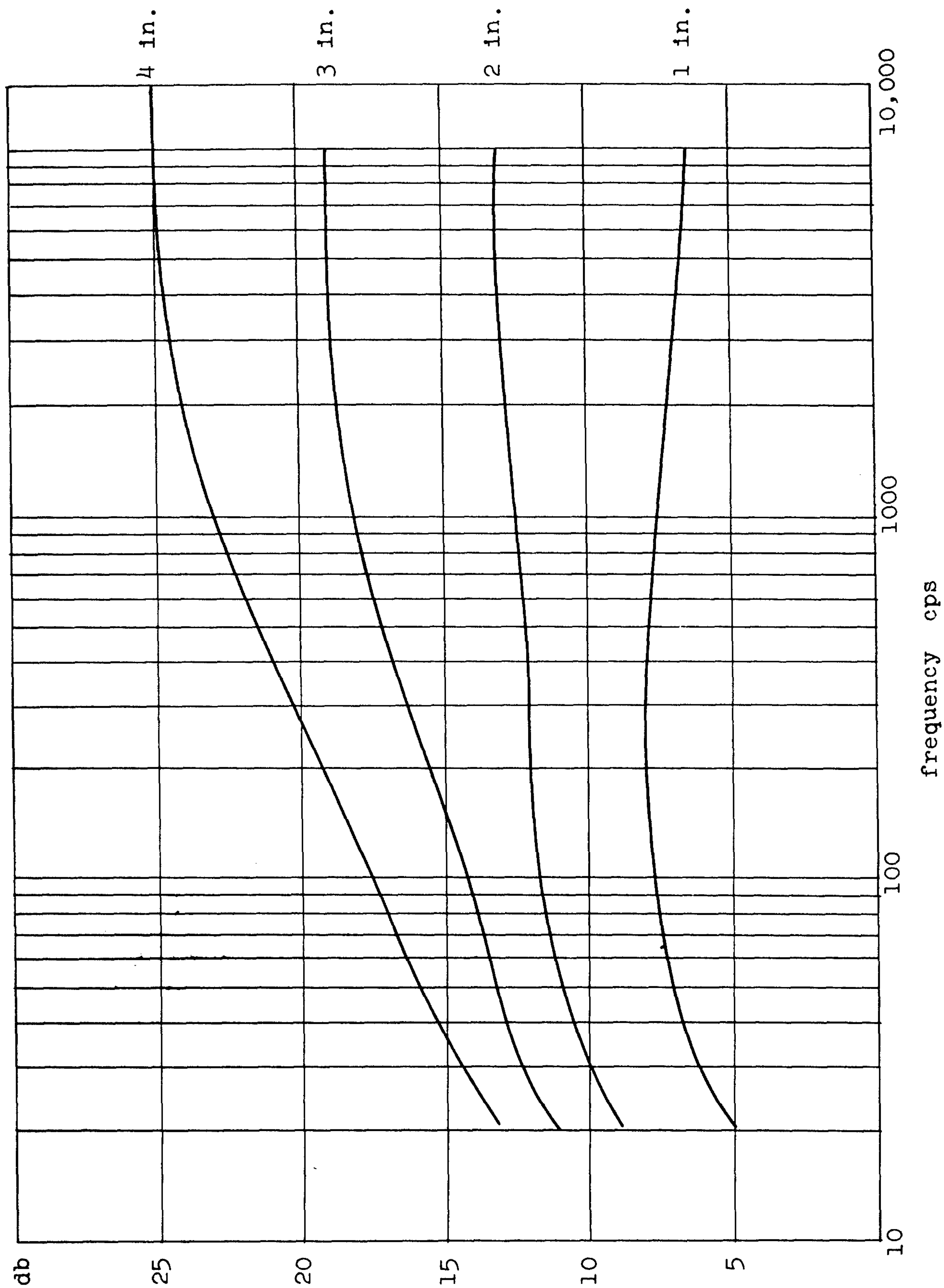


Figure 11.18

Equivalent circuit for wrapping alone which is valid at low frequencies.

Figure 11.19

Noise reduction of various thicknesses of 6 lb/ft^3 PF Fiberglas alone.



4 in.
3 in.
2 in.
1 in.
db
25
20
15
10
5
10
100
1000
10,000
frequency cps

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Figure 11.20

Noise reduction of various thicknesses of 10.5 lb/ft^3 PF Fiberglas alone.

CHAPTER 12

CONTROL OF AIRBORNE NOISE

12.1 Introduction

This chapter contains design and performance information for a number of sound-attenuating structures or principles which are applicable in the control of airborne noise. The topics included are the following:

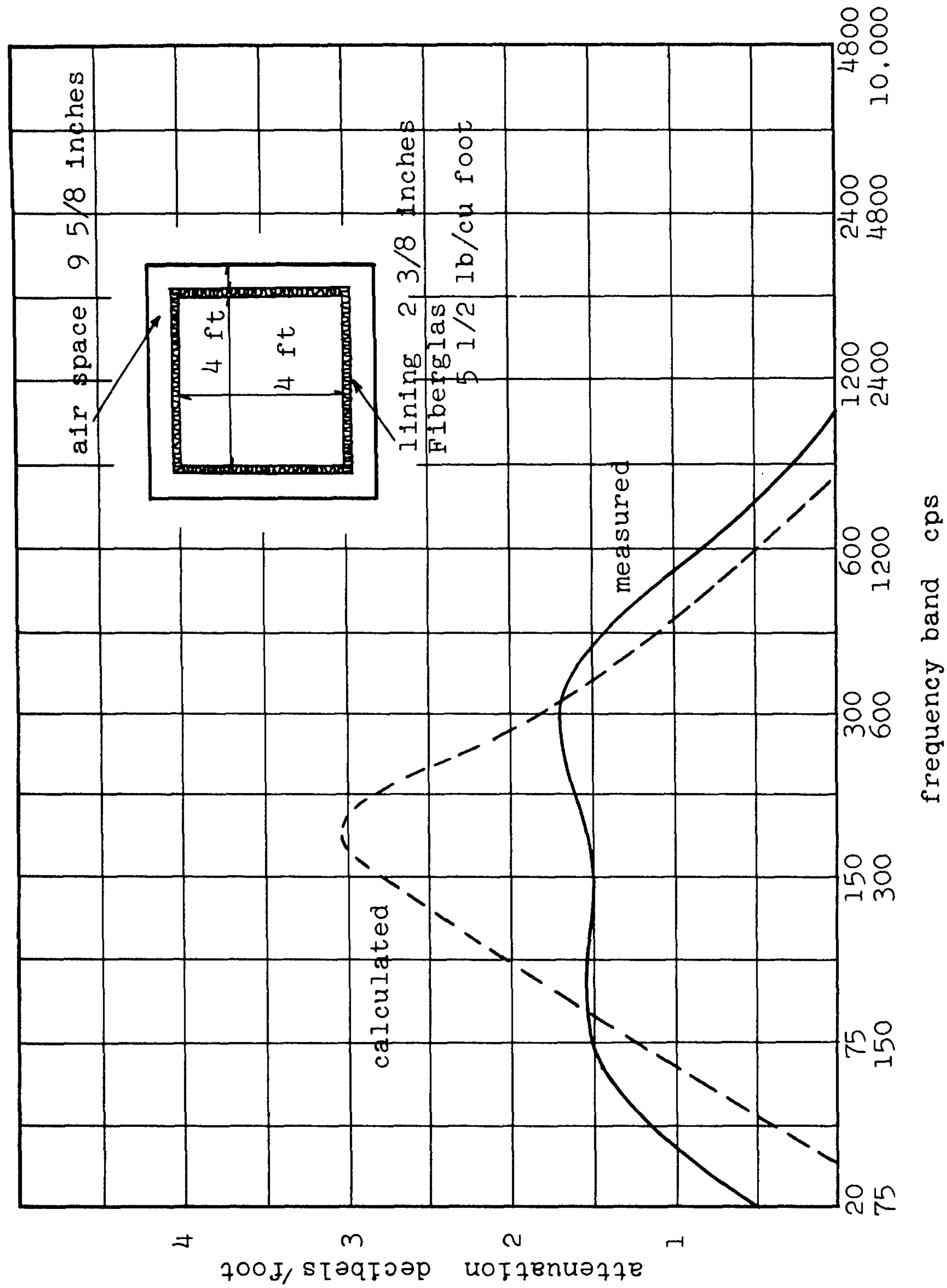
Ducts with absorbent lining
Parallel baffles
Duct bends
The duct with attached resonator
The duct with resonant lining
Sound propagation in the atmosphere
The isolation wall for outdoor sound control
Directionality of radiation
Combined treatments
Installation precautions.

12.2 Lined Ducts

The term "duct" will be taken to mean a channel, of uniform cross-section, which is bounded by rigid, impervious walls. A straight duct, when partially or fully lined with an acoustical absorbing material of suitable specifications forms a band-elimination filter. At one or more frequencies, the attenuation constant reaches a maximum value. The frequency at which maximum attenuation occurs is affected by the width of the duct and by the properties of the absorbent lining. In most practical cases, the condition of the maximum attenuation is realized when the wavelength of sound has a value which is between the width dimension of the duct and twice this dimension.

In the case of a rectangular duct, the width is measured perpendicular to a pair of lined walls; while in the case of a circular duct, the diameter may be taken as the effective width.

The duct lining, which usually consists of a porous material such as rock wool or Fiberglas, may be separated by an air space from the rigid outer wall. Such an arrangement is an economical method for increasing the effective depth of a given amount of acoustical material. Increasing the effective depth of the lining lowers the frequency at which maximum absorption is secured. The air space behind the material must be partitioned so as to prevent direct transmission of sound along the


air space 9 5/8 inches
lining 2 3/8 inches
Fiberglas 5 1/2 lb/cu foot
4 ft
4 ft
measured
calculated
attenuation decibels/foot
1
2
3
4
frequency band cps
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10.000

length of the duct. The region of highest attenuation can be broadened by suitable choice of air space depth, lining thickness and partition spacing. The problems involved here have not been examined in a completely general way and will not be treated here.

A fair approximation to the acoustical performance of lined ducts can be obtained from calculations based on the theory developed by Morse. 1,2/ Some discrepancy has usually been found between calculated and observed performance. In general the width of the attenuation peak is greater than the predicted value, and often the calculated value of maximum attenuation is not achieved. In practice, maximum attenuations along a duct of 7 to 9 db per unit width can be obtained at frequencies below 400 cps. To achieve these values of attenuation, the duct must be a minimum of one wavelength long at the lowest frequency for which useful attenuation is desired.

Results for Rectangular Ducts. Figure 12.1 shows a comparison between calculated and observed attenuation values for a particular duct. These measured values are derived from plots of octave-band sound levels as a function of distance through the duct. Specifications of the duct are given in the figure.

Experimentally observed results for a number of other cases are shown in Fig. 12.2. The attenuation values shown in this figure have been reduced to decibels for a length equal to the width, since this method of expressing the results gives values of the same order of magnitude for ducts of different sizes and is readily adaptable to scaling. Design data and measurements information for the ducts of Fig. 12.2 are given in Table 12.1. Case F duplicates the experimental case of Fig. 12.1. For the technique used to obtain the experimental results, see Sec. 14.2, method 4.

The most satisfactory engineering practice in predicting duct performance is to work from results obtained experimentally provided that the engineering problem requires a duct geometrically similar to one whose performance has been measured. Attenuation per unit width of duct length is substantially unaffected by linear scaling of dimensions (width dimensions, lining thickness, air space thickness) provided that the material of the lining is so chosen as to keep the specific

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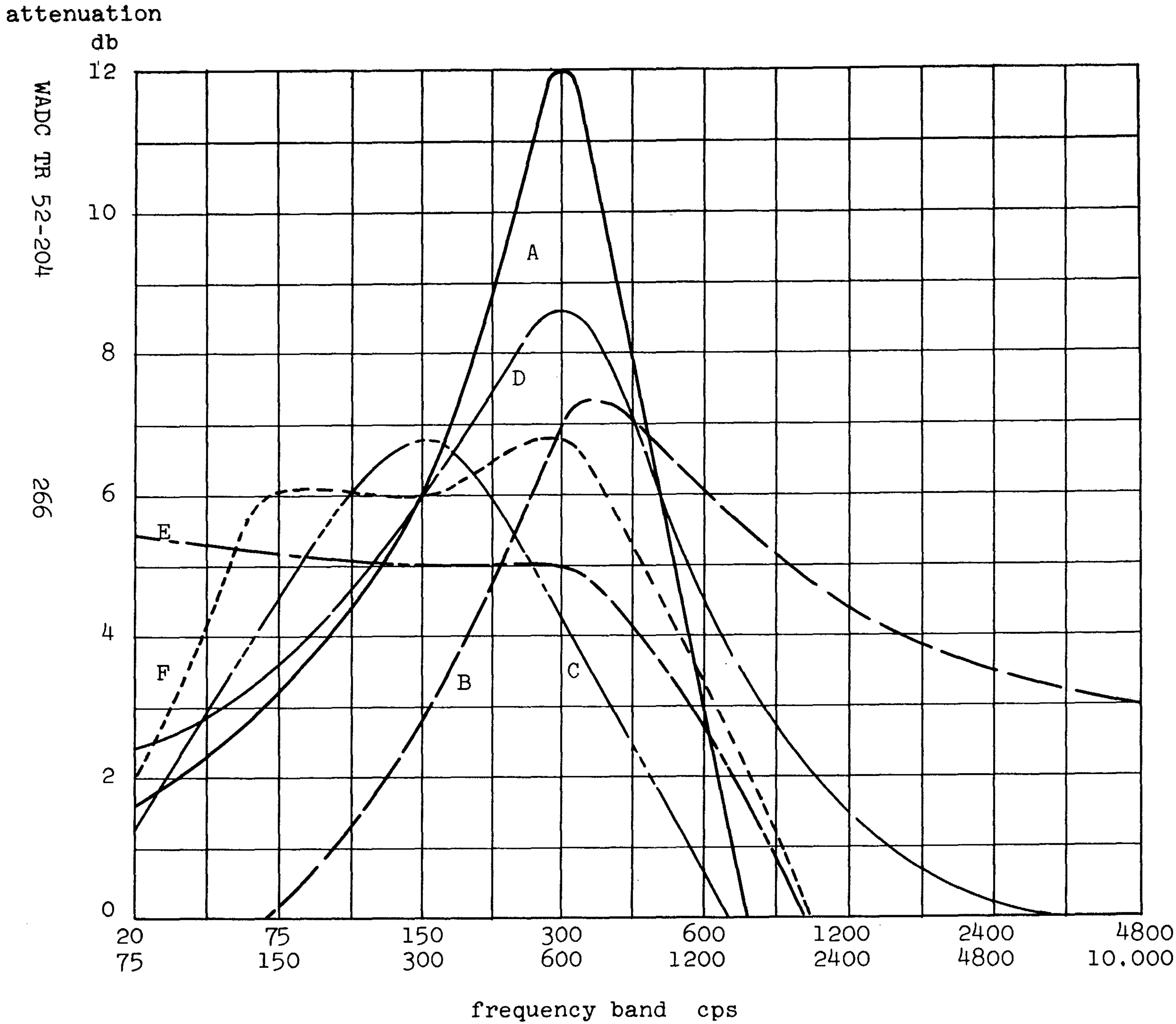
Figure 12.1

Comparison between computed and measured values of the attenuation in db/ft of a lined duct.


attenuation
db
12
10
8
6
4
2
0
A
B
C
D
E
F
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10.000
frequency band cps

flow resistance of the lining a constant. Figure 12.3 shows the flow resistance per inch of thickness for several lining materials as a function of the volume density.

The frequency of maximum attenuation varies in the scaling process inversely as any linear duct dimension; hence, as the size of the duct is increased, the plot of attenuation vs logarithm of frequency (or the plot of attenuation vs octave band) must be moved uniformly to lower frequencies. Curves E and F of Fig. 12.2 illustrate the approximate agreement obtained in results for ducts differing only in scale factor. The dimensions of duct E (10 ft square) are so large that the low-frequency decline of the attenuation is not apparent on the standard octave-band frequency plot.

Results for Circular Ducts. Prefabricated circular ducts with absorbent linings, usually constructed with heavy steel shells, are available commercially under the generic term "mufflers". Other structures which are commercially designated as "mufflers" and which include resonators as well as absorbent linings, will not be discussed specifically. The measured attenuation for several commercial mufflers (Maxim and Industrial Sound Control) is shown in Fig. 12.4. Physical data and measurements for these mufflers are shown in Table 12.2.

12.3 Parallel Baffles

In some cases the frequency of maximum attenuation which can be reasonably achieved by the use of duct linings may be too low. One means of obtaining attenuation at higher frequencies is to install rigid longitudinal partitions which effectively subdivide the original duct into smaller ducts. If the partition spacing is suitably chosen, the sub-ducts will, when properly lined, have maximum attenuation in the required higher frequency range.

Another solution, which may be less complicated structurally, is to subdivide the duct with a series of baffles of acoustically absorbing material which are parallel to each other and to the duct axis. These baffles have no rigid core and

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Figure 12.2

Attenuation as a function of frequency (Octave-band) for the lined duct structures of Table 12.1 The vertical scale gives attenuation in decibels for a length of duct equal to the width.

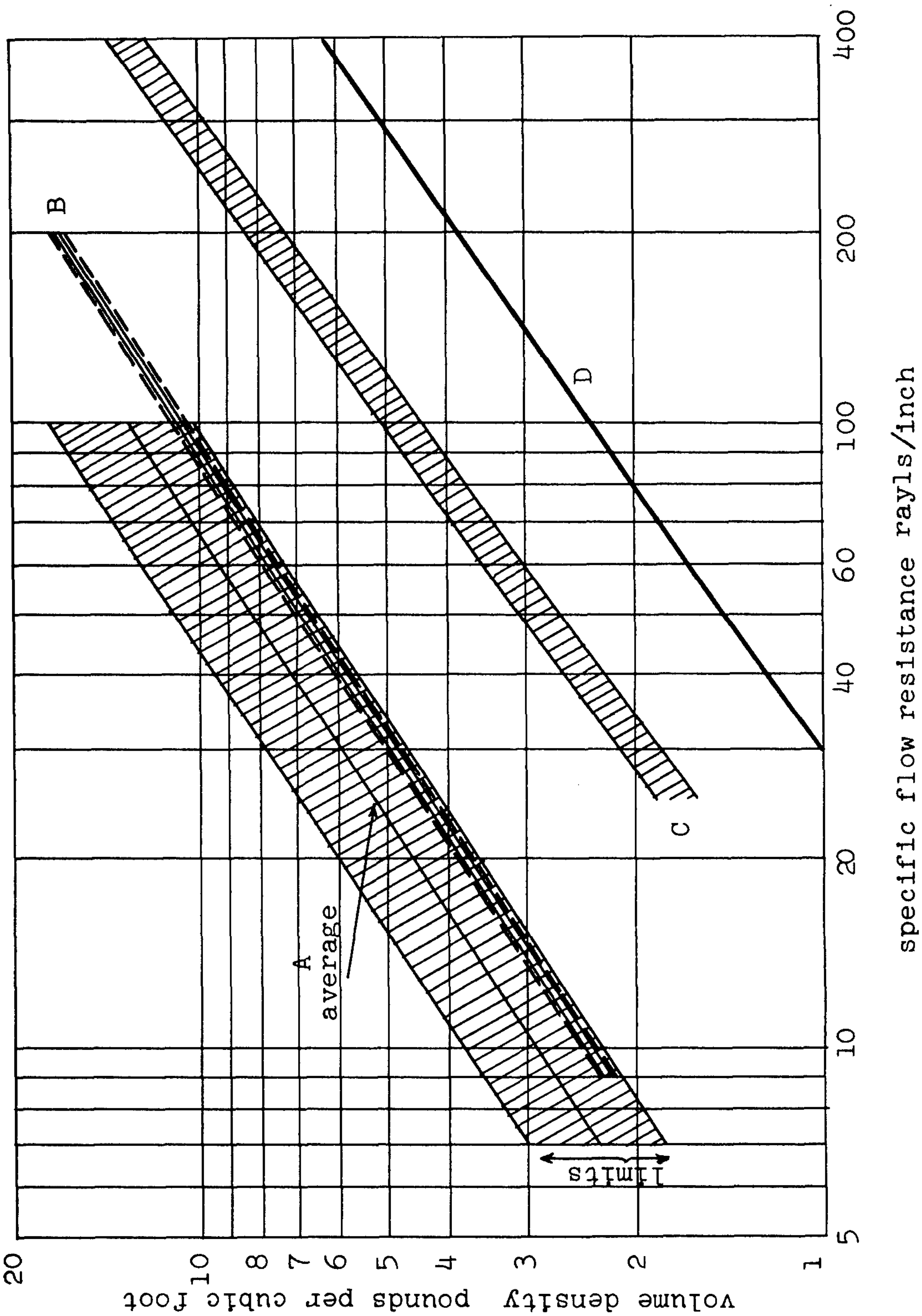

A
average
limits
B
C
D
specific flow resistance rayls/inch
volume density pounds per cubic foot
5
10
20
40
60
100
200
400
1
2
3
4
5
6
7
8
10
20

TABLE 12.1

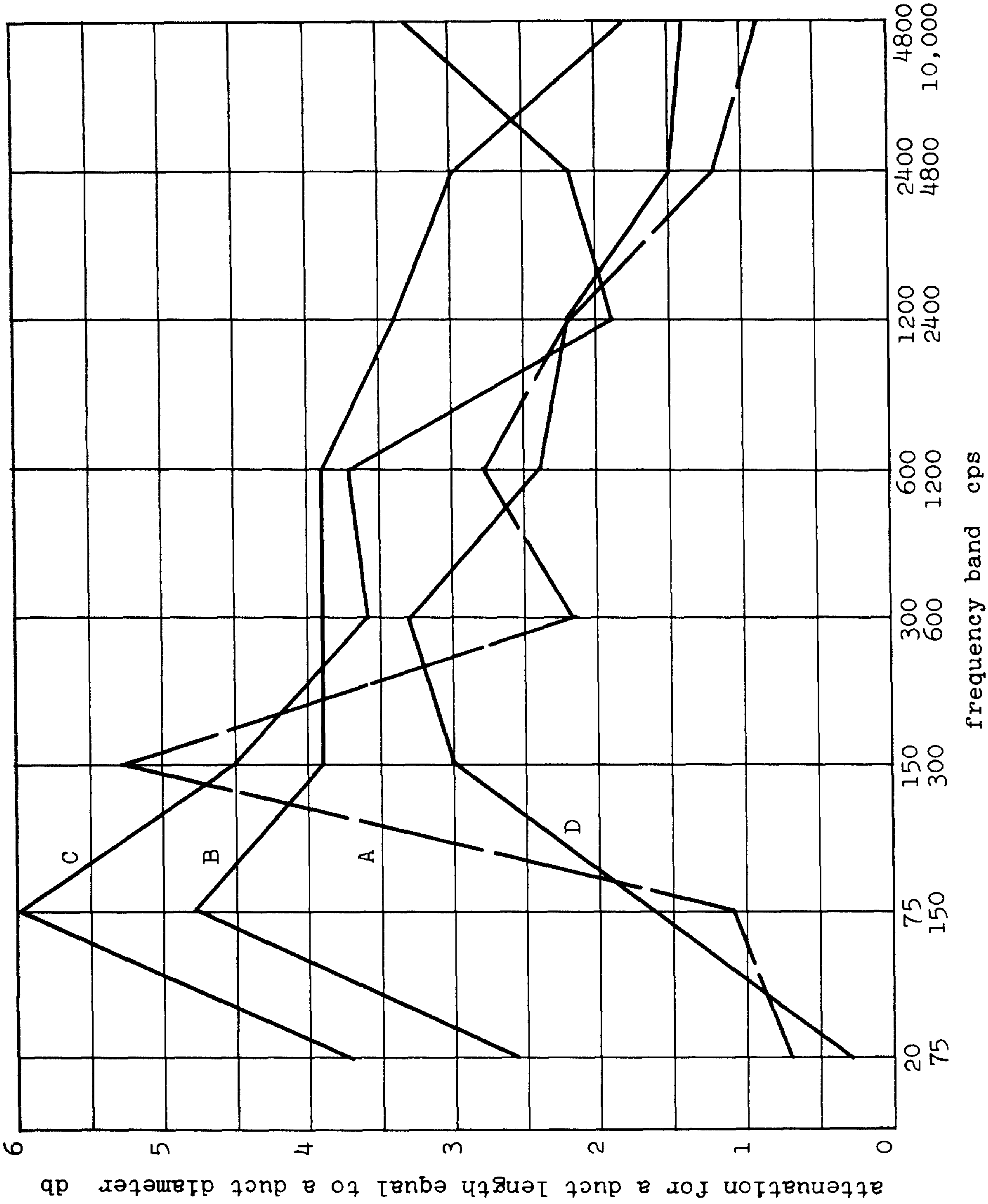
INFORMATION ON DUCTS FOR WHICH
ATTENUATION IS GIVEN IN FIG. 12.2
MINIMUM LENGTHS, 10 FT. MICROPHONE PLACED AT
INTERVALS OF 2 FT FOR ATTENUATION MEASUREMENTS

DUCT	A	B	C	D	E	F
Dimensions of Open Areas, ft.	3.8x12	2.3x30	2.3x30	3x30	10x10	4x4
Number of Sides Lined	2	2	2	2	4	4
Lining Thickness, Inches	16	4	6	6	6	2.4
Lining Material*	2.5#PF	6#PF	3#PF	3.5#PF	3#PF	5.5#PF
Depth of Air Space Behind Lining, in.	0	0	12	16	24	9.6
Frequency Bands for Measurement	1/3-Octave	Octave	Octave	Octave	Pure Tone	Pure Tone
Sound Source	Jet Engine	Reciprocating Engine			Pure Tone	Pure Tone

* Linings of PF Fiberglas, figures give density in lb/ft^3

Figure 12.3

Flow resistance in rayls, per inch of thickness, plotted as a function of volume density for several acoustical materials.



attenuation for a duct length equal to a duct diameter db
0
1
2
3
4
5
6
frequency band cps
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
A
B
C
D

TABLE 12.2

INFORMATION ON COMMERCIAL MUFFLERS
FOR WHICH ATTENUATION IS GIVEN IN FIG. 12.4
ALL LENGTHS 15 FT OR GREATER

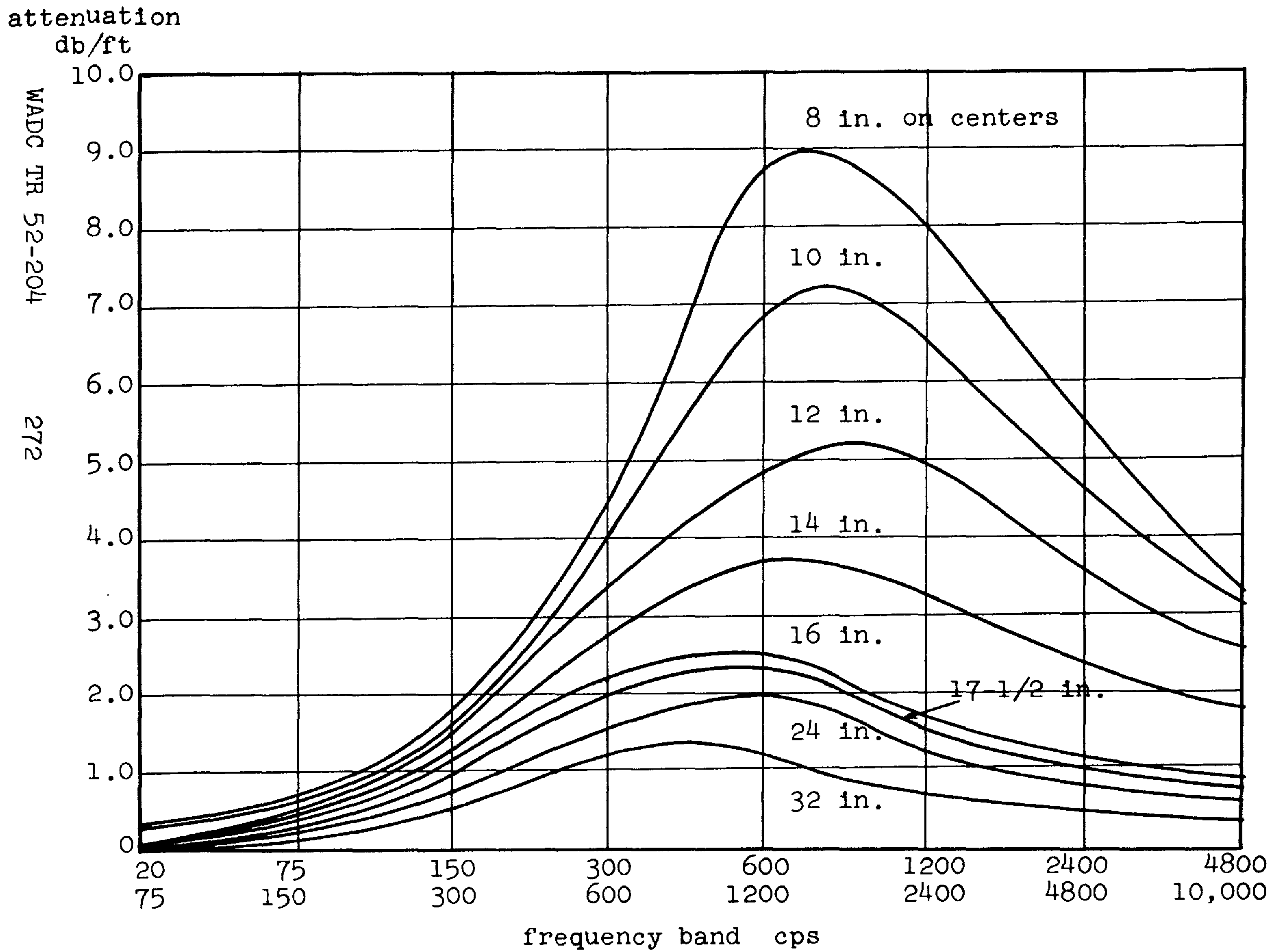
MUFFLER	A	B	C	D
Inside Diameter, in.	22	36	72	36
Lining Thickness, in.	3.5	3.4	5.3	(a) 2 in. (b) 4 in.
Lining Material	Copper Wool	Copper Wool	Copper Wool	(a) Monoblock (b) JM-305 PF Blanket
Air Space Behind Lining, in.	13	17.5	20	0

(a) Layer next to air stream
(b) Layer between a and shell

therefore the spaces between these baffles are not ducts as defined in Sec. 12.2. The acoustical performance of the parallel baffle structure is, however, qualitatively similar to that which would be obtained by subdividing the original duct into smaller lined ducts. The performance of the parallel-baffle structure is not quantitatively the same as that for a similar structure with rigid-core partitions, on account of acoustic coupling between adjacent channels which is present when the partitions are porous.

Figure 12.4

Attenuation as a function of frequency (Octave-band) for the commercial mufflers of Table 12.2


attenuation
db/ft
10.0
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0
8 in. on centers
10 in.
12 in.
14 in.
16 in.
17-1/2 in.
24 in.
32 in.
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

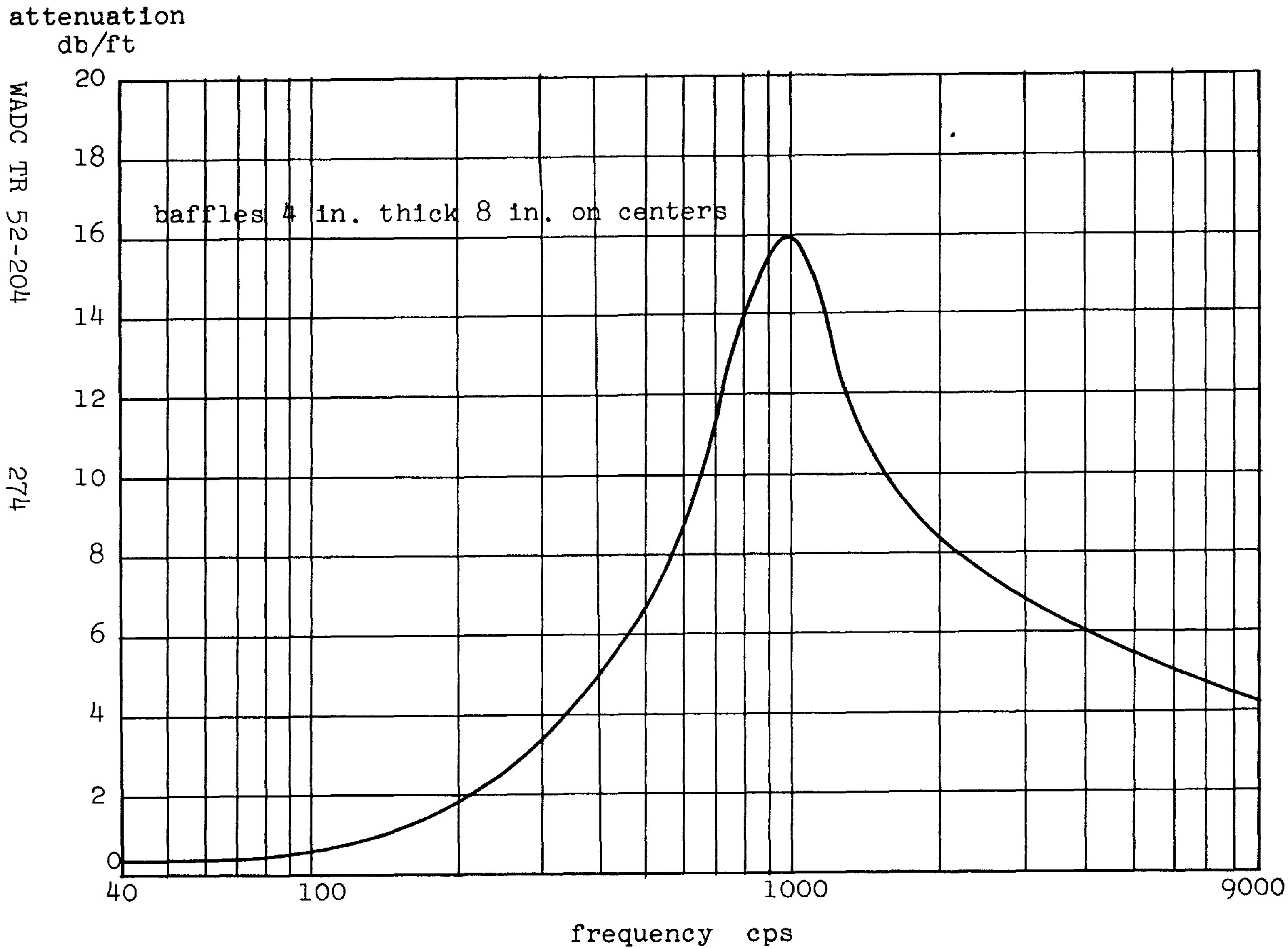
No general theory is available for the performance of parallel-baffle structures, and engineering experience with these structures is not sufficiently extensive to lead to general design charts. Experience with a particular family of structures in which the baffle spacing is varied, while the structure of the individual baffles remains constant, is summarized by the attenuation curves of Fig. 12.5. The results as shown apply to the case of a continuous-spectrum noise source. Two of the curves (those for 10 in. and 14 in. on-center spacing) were obtained by interpolation between the curves to which they are adjacent. The others represent experimental measurements performed by the techniques used for ducts, with pure-tone signals.

While each attenuation curve of Fig. 12.5 has a maximum with respect to frequency, the results differ in other features from those for ducts. The frequency of maximum attenuation changes relatively little as the spacing is increased. This appears to indicate that the frequency of maximum absorption is determined primarily by the acoustic properties (particularly the thickness) of the baffles. As in the case of the lined duct the attenuation decreases rapidly below the frequency of maximum loss. The attenuation at frequencies above the peak, however, is somewhat greater in the parallel-baffle structure than in the duct.

A more detailed presentation of the attenuation function for one of the parallel-baffle structures is given in Fig. 12.6, which shows directly the measured attenuation as a function of frequency instead of the effective octave-band values. This curve, which is for 8-in. center-to-center spacing, shows the original data from which the top curve of Fig. 12.5 was calculated. While the attenuation constant rises to a sharp peak of 16 db/ft at 1000 cps, this value is useful only if a discrete-frequency sound near 1000 cps must be eliminated. When the overall energy reduction for an octave band is considered, the largest effective value is 9 db/ft as indicated in Fig. 12.5

Figure 12.5

Attenuation as a function of frequency (octave-band) for a family of parallel baffles in which the spacing is varied. In all cases each baffle is 4 in. thick and filled with either PF Fiberglas of density 4 lb/ft^3 or Rockwool of density 6 lb/ft^3. The results as shown apply for continuous-spectrum noise.


attenuation
db/ft
20
18
16
14
12
10
8
6
4
2
0
baffles 4 in. thick 8 in. on centers
40
100
1000
9000
frequency cps

A structure which is in effect two mutually perpendicular sets of parallel baffles can be formed by filling a duct or an opening with porous-concrete building blocks. The blocks are installed with the individual openings in successive layers aligned to form continuous ducts, parallel to the axis of the containing duct. The porous concrete block used for this purpose is sold under such names as Celocrete, Haydite, and Soundstone.

While the porous-block treatment produces somewhat less attenuation per foot than mineral wool blankets of comparable spacing, the treatment in some applications has vibration-reducing or resisting properties not afforded by acoustical blankets. The porous-block treatment has been used, for example, to prevent destructive vibration of the containing walls in the case of ducts carrying high-energy air streams, and at the same time to afford useful noise reduction.

The measured attenuation for an 18 ft length of Soundstone treatment, is shown in Table 12.3. The structural arrangement is shown in Fig. 12.7 (a,b). The reason for inclining the channels is given in Sec. 12.4.

12.4 Duct Bends

The duct bend is a simple and useful structure for adding high-frequency attenuation to a duct system. In intermediate frequency ranges the bend is an alternative or supplement to parallel baffles or subdivided ducts. In the highest frequency ranges the duct bend or some equivalent is often the only available means of introducing large attenuation, for in general it is not practical to decrease the spacing of duct partitions to a sufficiently small value to produce peak duct attenuation at frequencies as high as 5 to 10 kc; similarly the maximum absorption for parallel baffles cannot conveniently be placed in this frequency range.

A duct bend treated with absorbing material is somewhat more effective than an unlined bend. The most common arrangement

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Figure 12.6

Attenuation, as a function of frequency in cps, for parallel baffles with 8 in. center to center spacing. The values shown by this curve were used to derive the top curve of Fig. 12.5

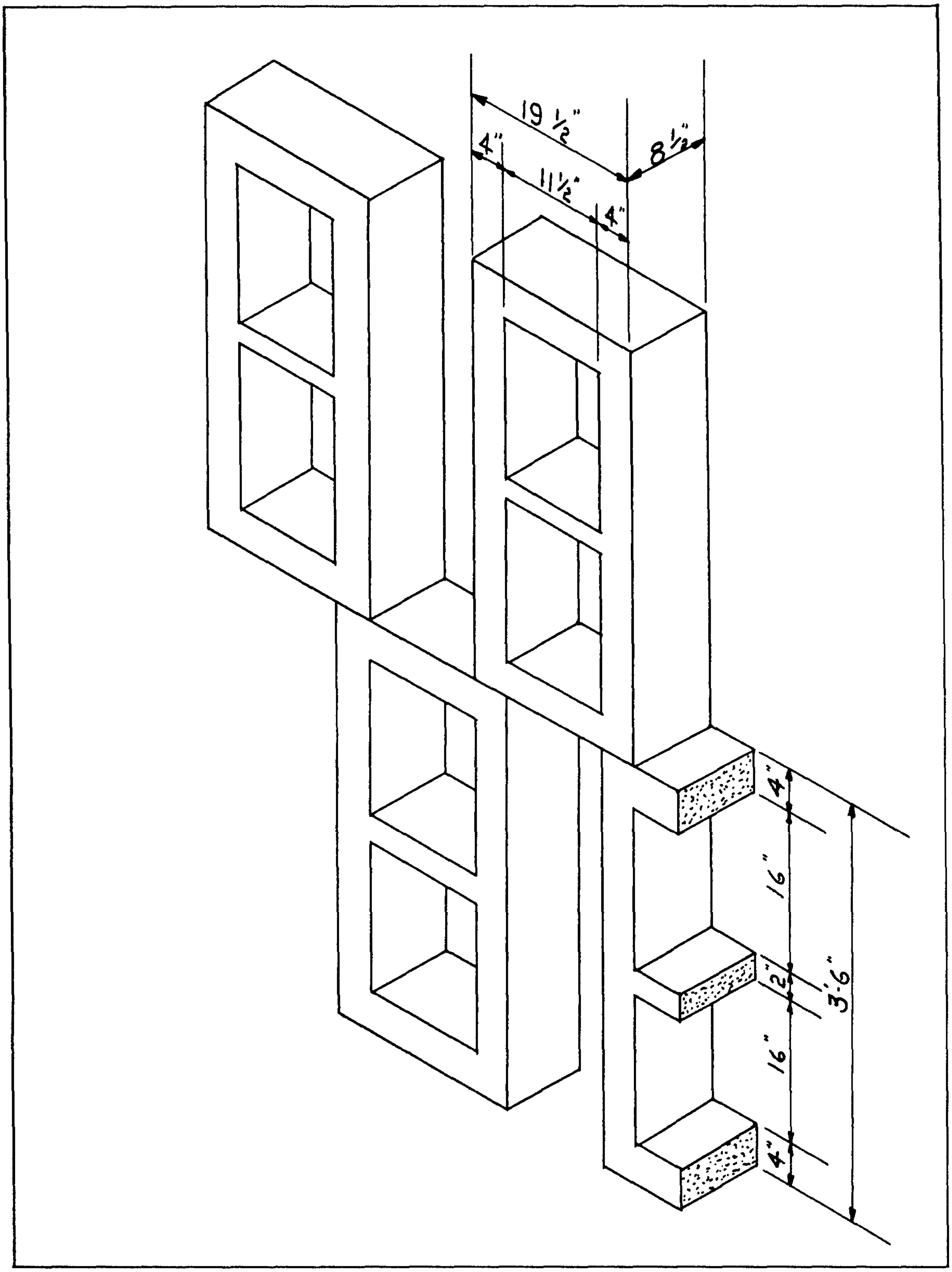

19 1/2"
8 1/2"
4"
11 1/2"
4"
4"
16"
2"
3'6"
16"
4"

Figure 12.7 (a)

Soundstone blocks and the attenuating structure for which data are given in Table 12.3

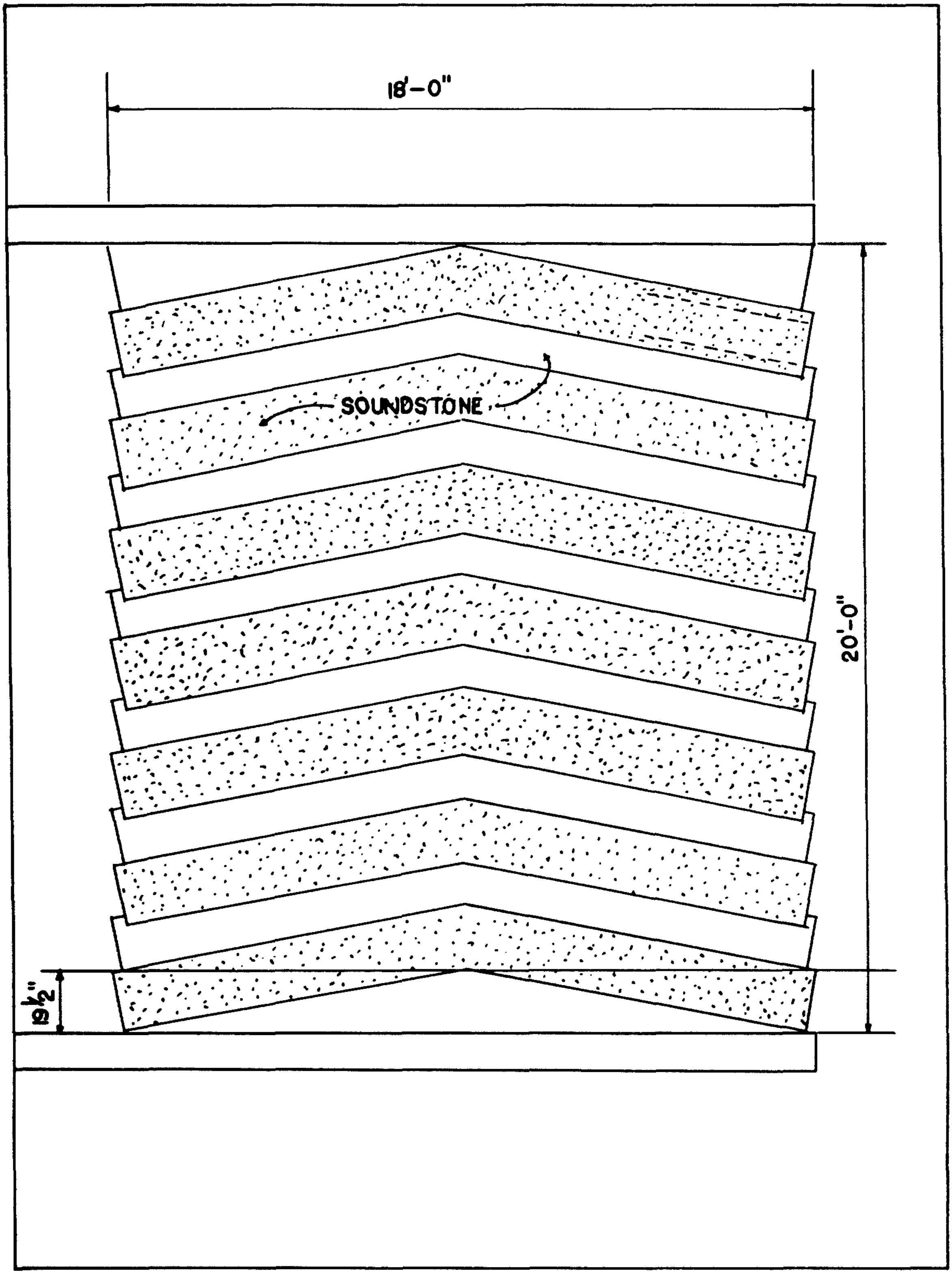

18'-0"
SOUNDSTONE
20'-0"
19½"

TABLE 12.3

MEASURED ATTENUATION OF A SOUNDSTONE INSTALLATION
LENGTH 18 FT

Frequency Band	Total Attenuation	Attenuation in db/ft
cps	db	
20-75	2	0.11
75-150	7	0.39
150-300	13	0.72
300-600	26	1.45
600-1200	40	2.2
1200-2400	36	2.0
2400-4800	39	2.2
4800-10000	36	2.0

is to line the end wall of the bend as shown in Fig. 12.8. The action of the bend may be explained qualitatively by the statement that incoming sound waves travel across the bend to strike the absorbent lining, where a large portion, reflected back toward the source, is partially absorbed upon again traversing the incoming duct section. A relatively small portion of the sound energy is diffracted in such a way as to travel down the portion of duct beyond the bend. Even if no lining is used at the bend, considerable attenuation results from the reflection of sound energy back toward the source. It can be shown that this explanation is not valid when the wave-length of the sound is much greater than the width of the duct; therefore no appreciable attenuation is introduced at such low frequencies.

Figure 12.7 (b)

Soundstone blocks and the attenuating structure for which data are given in Table 12.3

On the basis of measurements on laboratory scale models (duct 8 x 8 in) and on a large installation (duct 20 x 20 ft), an approximate design curve has been developed for the attenuation due to a lined duct bend of the type shown in Fig. 12.8. This design curve is plotted on a frequency scale and on an octave-band scale in Figs. 12.9 and 12.10 respectively. Geometrically similar ducts should yield identical results for attenuation as a function of a dimensionless frequency parameter, provided the acoustic impedance of the lining as a function of this same frequency parameter is kept constant. The frequency parameter is d/λ or df/c, where $\underline{d}$ is the transverse dimension of the duct (Fig. 12.8), $\underline{\lambda}$ is the wavelength of the sound in open air, $\underline{f}$ is frequency and $\underline{c}$ is the speed of sound in air. The design curve has been adapted on this basis to various widths, as shown in Figs. 12.9 and 12.10. The calculations are based on the value 1120 ft/sec for the speed of sound.

The outstanding feature of the results is that the bend loss rises rapidly as the frequency increases above the lower limit of the first cross-mode of the duct. This frequency limit corresponds to $df/c = 0.5$.

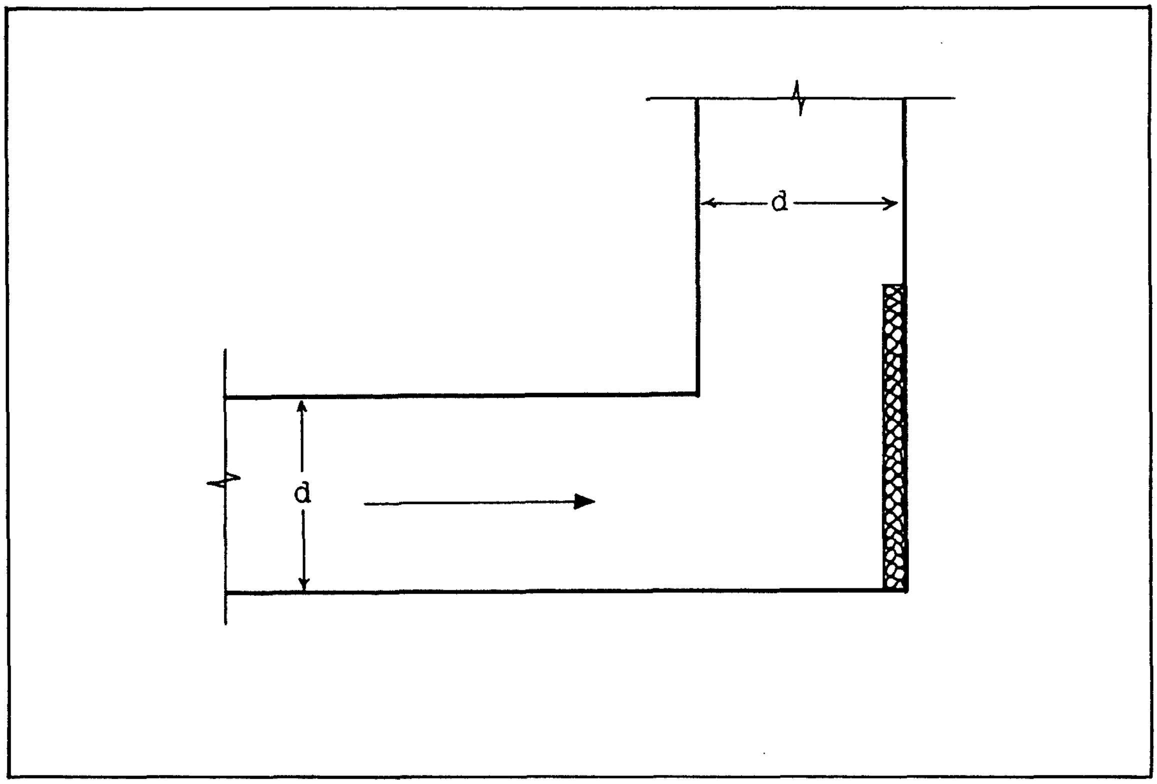
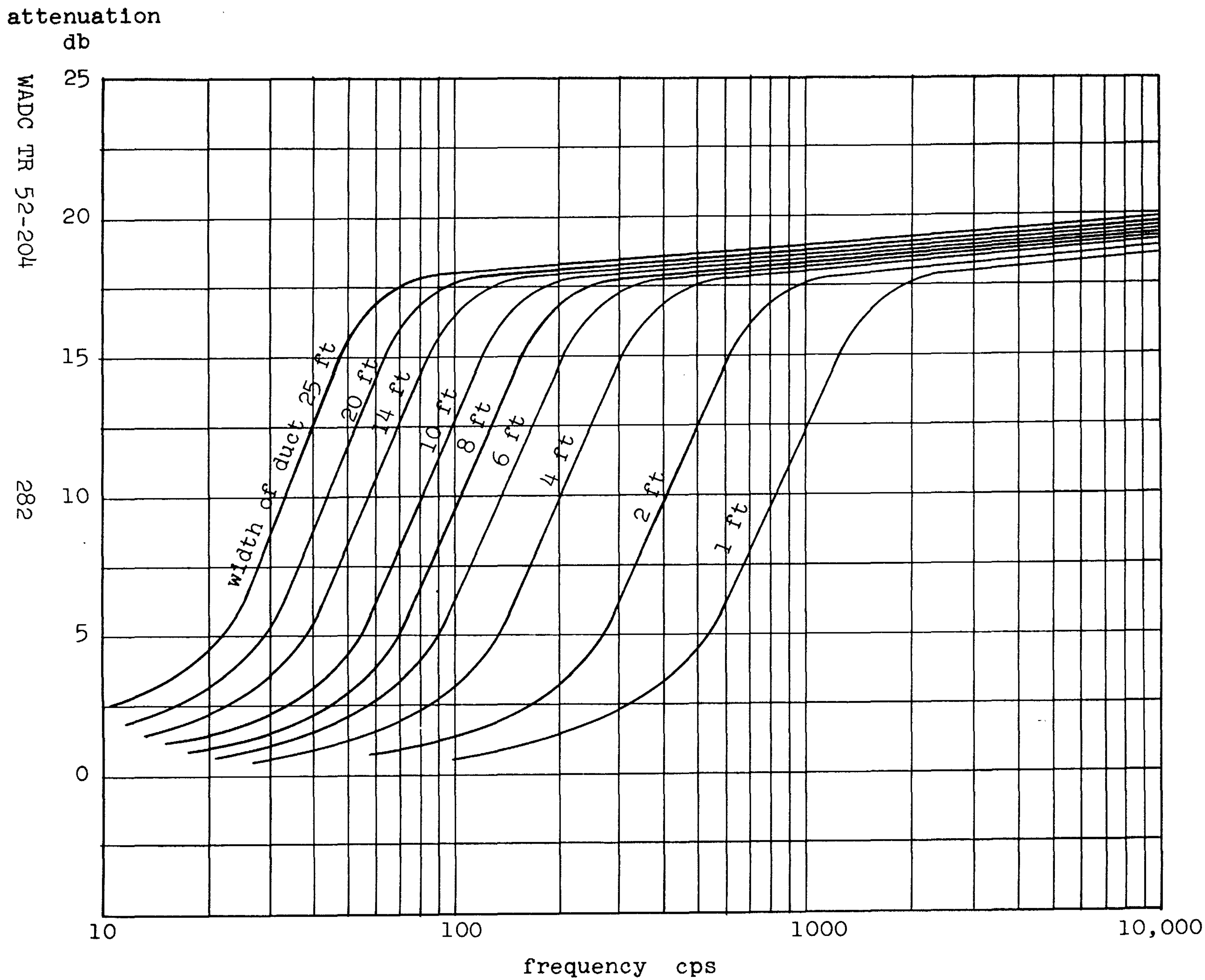


Figure 12.8

Sketch showing a lined duct bend.


attenuation
db
25
20
15
10
5
0
width of duct 25 ft
20 ft
14 ft
10 ft
8 ft
6 ft
4 ft
2 ft
1 ft
10
100
1000
10,000
frequency cps

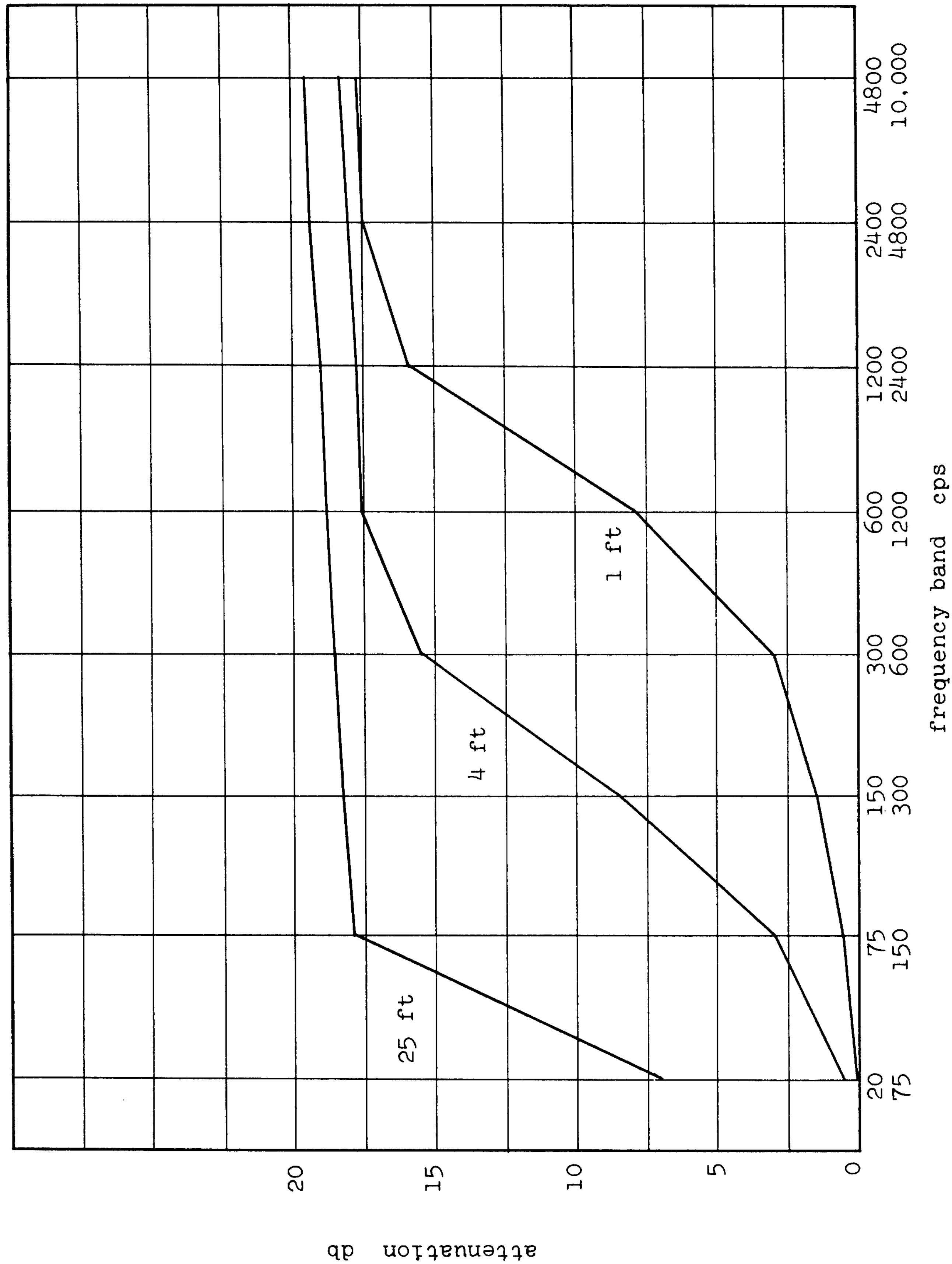
The design curve is rounded off to a maximum loss of slightly less than 20 db. This is a conservative value which allows for the presence of untreated edges which will produce some scattering at high frequencies. If such scattering is substantially eliminated by a complete acoustic lining, which covers all sides of the duct for a distance beyond the bend of approximately one duct width, the limiting high-frequency attenuation will be at least 5-10 db above that shown as the upper limit by the design curve.

According to the experimental data on which the design curve is based, the loss at the bend is not critical with respect to the thickness of the lining. For all linings used in the experimental measurements, however, the normal absorption coefficient had values in excess of 0.8 at a frequency given by df/c = 0.5; thus the independence of loss and lining thickness is not a completely general conclusion. It is suggested that this design curve be used only in cases where the absorption coefficient is 0.8 or more at all frequencies for which df/c is greater than 1.5. For practical lining materials this requirement means that the lining thickness should be of the order of d/10 or greater.

An arrangement which is in some respects similar to the lined bend, but much less effective, is the use of sets of parallel baffles, or lined partitions, whose planes are inclined at a few degrees to the duct axis. If this treatment is continued for a length of several duct widths, it becomes necessary to use a zig-zag arrangement, that is, successive sets of baffles inclined in opposite directions to the duct axis. One general objective of bend treatments, sloping parallel baffles, and other treatments on this class is to eliminate line-of-sight propagation of sound at high frequencies and in this way to insure that high-frequency sound must strike one or more surfaces at which the sound energy may be in part absorbed and in part reflected to other absorbing surfaces. Line-of-sight propagation is not a problem in the lower frequency range, where the wavelength is of the order of the duct width or greater; here the sound energy diffracts readily into the absorbing linings even in straight ducts. If it were possible to provide a sufficiently thick lining for a straight duct, a large attenuation could be secured at any low frequency, but a bend or equivalent treatment would still be required at high frequencies.

Figure 12.9

Attenuation as a function of frequency in cps for a lined bend in ducts of various widths.


25 ft
4 ft
1 ft
20
15
10
5
0
attenuation db
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps

12.5 The Waterspray Muffler

The waterspray muffler is a highly specialized acoustical attenuator which has arisen from the problems peculiar to the quieting of jet engine test facilities. These facilities usually include mufflers to reduce external noise levels. The high temperature of jet engine exhaust gases makes some sort of cooling desirable in order to prevent damage to the exhaust muffler. One cooling scheme is to spray water directly into the exhaust stream before the gas enters the muffler. One way of doing this is by means of many small nozzles distributed around the inner surface of the exhaust pipe as shown in Fig. 12.11. Under certain conditions the waterspray may cause some attenuation of the sound. In some cases, the waterspray alone may effect the necessary noise reduction without any additional muffling. However, it may prove that the greatest economy in achieving a desired overall sound attenuation will be obtained by a combination of waterspray and conventional muffler.

The sound attenuation in a waterspray muffler is due to viscous losses in the air.* Because of the repeated reflections inside the exhaust at the higher frequencies, the effective sound path is lengthened and these viscous losses operate over a greater distance. The attenuation in a waterspray muffler depends indirectly on the amount of water in the air, the density of the air, the viscosity, etc. Consideration of the thermodynamic equilibrium inside the muffler indicates that the muffler

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* Relaxation effects, due to conversion of sound energy to molecular vibrations, may be expected to cause further "anomalous" absorption. The air is saturated with moisture over the entire length of the muffler. Since the relaxation frequency for saturated air is in the ultrasonic range, such effects are less significant at audio frequencies. (See Ref. 3.)

Figure 12.10

Attenuation as a function of frequency (octave-band) for a lined bend in ducts of various widths. Derived from data of Fig. 12.9 for continuous-spectrum noise.

can be divided into two regions. In the first zone, because of the high temperature of the incoming exhaust gases, the rate of evaporation is quite high and the moisture content of the air increases continuously until saturation is reached. Over that region the vapor is superheated. In the second zone, more and more of the superheated vapor condenses to maintain the air always at its saturation point. The thermodynamic conditions are shown in Fig. 12.12. A calculation of the temperature distribution shows that except for a narrow region at the entrance of the muffler, the temperature is substantially uniform and equal to that of the cooling water. However, a theoretical analysis indicates that the attenuation of sound is strongly dependent on the size of the water drops. Since this cannot be calculated easily, the attenuation of a waterspray muffler cannot be predicted with certainty at the present time.

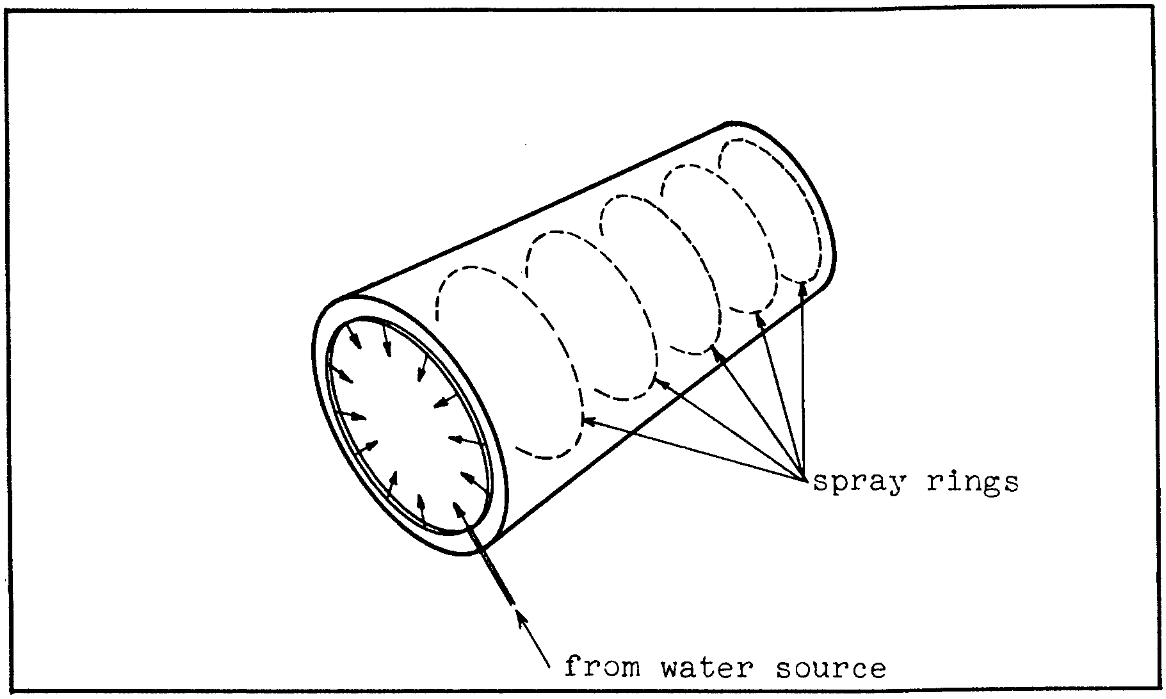


Figure 12.11

The waterspray muffler

12.6 Duct with Attached Resonator

The lined duct is not ordinarily suitable for providing large attenuation of those low-frequency sounds which have a wavelength many times greater than the diameter of the duct. The thickness of duct lining required to produce useful attenuation in this frequency range is too great to be practical. Large attenuation for a narrow range of low frequencies can, however, be obtained by installation of a resonator which opens into the duct wall. No duct lining is required. A number of different resonators (of the order of six or more per octave band of frequency) must be used to obtain essentially continuous attenuation over a wide range of low frequencies.

The action of the resonator may be explained with reference to Fig. 12.13. Figure 12.13a shows schematically the physical arrangement. An acoustic signal of rms pressure amplitude p_1 travels along the duct from the sound source toward the location of the resonator. At the resonator opening a portion of the acoustic disturbance is effectively diverted to produce motion of the air in the resonator. Accordingly the rms acoustic pressure p_2 in the portion of the duct beyond the resonator is smaller than the input pressure p_1.

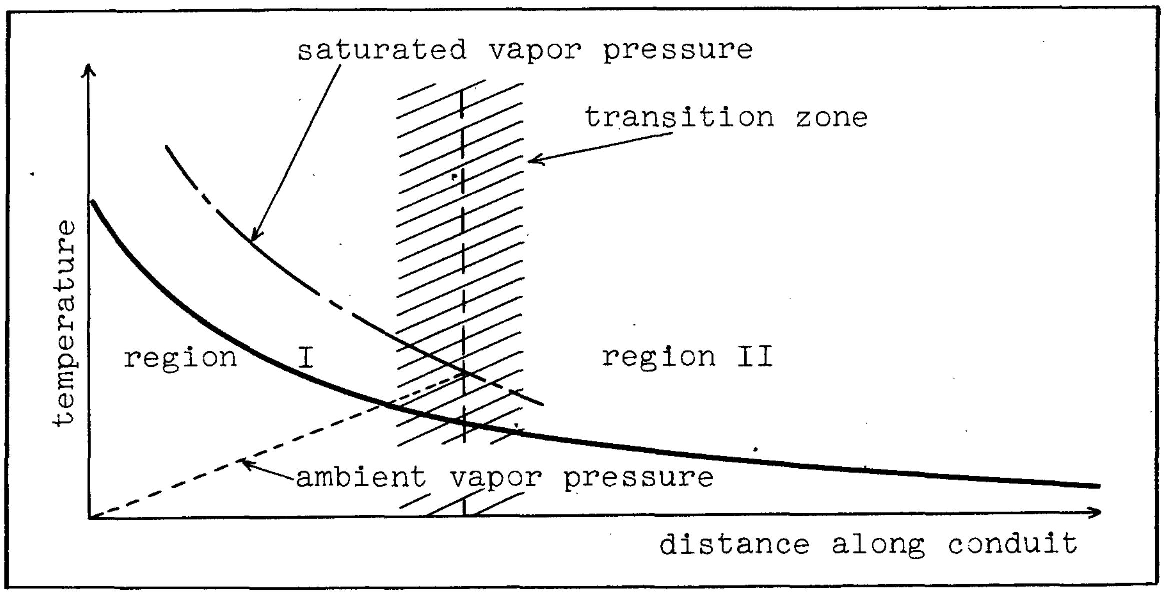


Figure 12.12

Sketch showing qualitative behavior of temperature and vapor pressure, in a waterspray muffler, as a function of distance from the entry.

A quantitative analysis can be given in terms of the equivalent circuit of Fig. 12.13b. The section of duct on the source side of the resonator can be represented approximately by an acoustic resistance $\rho c/S$, where ρ is the density in gm/cm^3 of the gas, c is the speed of sound in the duct in cm/sec (essentially equal to the velocity in open air at the same temperature), and S is the cross-sectional area of the duct in sq cm. A generator which would provide an open-circuit rms pressure in dynes/cm^2 of $2p_1$ is connected in series with this input section of the duct. The output pressure p_2 is developed across the acoustic impedance of the output section

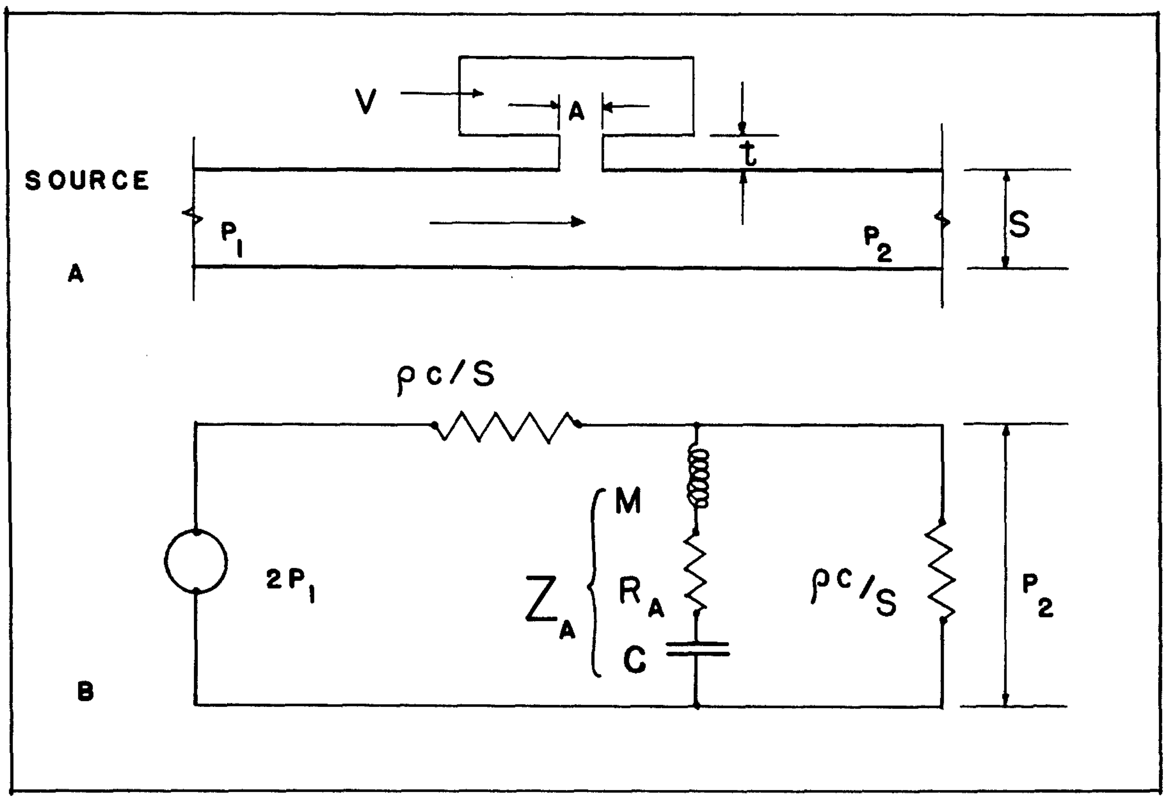


Figure 12.13

(a) Schematic diagram of a resonator attached to a duct.

(b) Analogous electrical circuit for the arrangement of (a).

of the duct. This impedance, in the absence of severe reflections in the output section, is approximately the resistance $\rho c/S$. Finally, as indicated in Fig. 12.13, the acoustic impedance Z_A of the resonator appears at the junction of source and load impedances. Solution of the circuit of Fig. 12.13b by standard methods reveals that the ratio of the output pressure with no resonator present, p_1, to p_2 the output pressure with the resonator present is given by

$$\frac{p_1}{p_2} = 1 + \frac{\rho c}{2 S Z_A} .$$

According to this equation, the output pressure decreases rapidly as Z_A becomes smaller than $\rho c/S$.

The impedance Z_A is approximately the series impedance of three circuit elements designated as M, C, and R_A in the figures. Therefore

$$Z_A = R_A + j \left(M\omega - \frac{1}{C\omega}\right)$$

where ω is the radian frequency. At the frequency of resonance, Z_A is equal to R_A.

The ratio p_1/p_2, which is a direct measure of the sound pressure reduction, is called the pressure ratio and is denoted by r. Subscripts o denote the condition of resonance, which is also the condition of maximum pressure ratio. Thus at the frequency of resonance, denoted by f_o, the pressure ratio is

$$r_o = 1 + \frac{\rho c}{2 S R_A} \qquad (12.3)$$

The symbol L is used for the attenuation in decibels, and is given by 20 log r. The resonance attenuation, 20 log r_o, is therefore denoted by L_o.

The acoustic mass M is given by the relation

$$M = \rho \frac{t + 0.96 \sqrt{A}}{A} \qquad (12.4)$$

where t is the length in cm of the communicating tube of the resonator and A is the cross-sectional area in sq cm of this tube.

The acoustic compliance C is given by

$$C = \frac{V}{1.4\ P_o} \tag{12.5}$$

where V is the volume in cubic cm of the resonator cavity and P_o is the barometric pressure in dynes/cm^2.

The acoustic resistance R_A, in the case of a circular opening, is given by

$$R_A = \frac{2\sqrt{\pi \mu \rho f}}{\pi a^2}\ [\ 2 + \frac{t}{a}] \tag{12.6a}$$

where a is the radius of the opening, μ is the viscosity of air, and the other terms are as defined previously. When the opening is not circular, approximately correct results are obtained from the slightly different form

$$R_A = \frac{P\sqrt{\pi \mu \rho\ \ t}}{A^2}\ [\ \frac{P}{\pi} + t] \tag{12.6b}$$

where P is the perimeter of the opening. When cgs units are specified and when the values of ρ and μ for air at 25°C, 760 mm barometric pressure* are inserted, the above two equations reduce to

$$R_A = \frac{5.28 \times 10^{-4}}{a^2}\ (\ 2 + \frac{t}{a})\ \sqrt{f} \tag{12.6c}$$

$$R_A = \frac{8.30 \times 10^{-4} P}{A^2}\ (\ \frac{P}{\pi} + t)\ \sqrt{f} \tag{12.6d}$$

---

* In the remainder of this section, the phrase "air under ordinary conditions" refers to air at approximately 25°C, and 760 mm pressure. Under these conditions $\rho = 1.18 \times 10^{-3}$ gm/cm3, $\mu = 1.8 \times 10^{-4}$ gm/cm sec. ρc = 41.1 rayls and the temperature coefficient of the viscosity coefficient is + .00494 x 10^{-4} gm/cm sec °C.

The frequency of resonance may be calculated from Eq. (12.7)

$$f_o = \frac{1}{2\pi\sqrt{MC}} \tag{12.7}$$

It is also desirable to have a measure of the width of the frequency range in which the attenuation remains near the resonance value, L_o. If the maximum attenuation is of the order of 20 decibels or more, then the following relation is closely correct.

$$\text{Frequency range over which attenuation L is within 3 db of maximum value } L_o = \frac{f_o}{Q} \tag{12.8}$$

The factor Q, which is a measure of the sharpness of resonance, is given by

$$Q = \frac{2\,\pi\,f_o\,M}{R_A} \tag{12.9}$$

Hence the frequency range (bandwidth) for which the attenuation is within 3 db of the maximum value is given by

$$\Delta f = \frac{R_A}{2\,\pi\,M} \tag{12.10}$$

In the foregoing discussion it is assumed that the resistance of the resonator results from viscous losses alone. Under certain extreme conditions, however, heat losses in the cavity may contribute significantly to the resistance. The contribution of thermal losses to the resonator resistance for air at 25°C, 760 mm pressure is given by

$$R_{thermal} = \frac{1.17 \times 10^4 A_c}{V^2 \; f_o^{3/2}} \qquad \text{(cgs units)}$$

where A_c is the area of the cavity walls. In the case of low-frequency resonators the thermal resistance is important only when the cavity has a shape which makes A_c unusually large in relation to V.

Resonator Design Information. While the preceding equations of Sec. 12.6 contain all necessary basic information on the properties of a resonator connected to a duct, it is not a straightforward matter to apply these equations without further

interpretation. A series of conclusions and simplified relations, drawn from the basic equations, are given below. These form the basis for an orderly design approach. In order to simplify the interpretation, it has been assumed that the resonator opening has a roughly circular or square shape, and that the thickness $\underline{t}$ of the orifice is so small in comparison to its width that the effect of the thickness on the impedance can be neglected. This condition is common in practice, as it is usually most convenient to allow a hole cut in the duct wall to function as the orifice. The frequency of resonance always appears as a parameter in the performance equations. This frequency is assumed to be held constant when the effects of other parameters are discussed.

1. The bandwidth for various degrees of attenuation less than the maximum value, L_o may be estimated from Table 12.4. Let L_{max} denote the maximum or resonant attenuation in db. It is desired to find the bandwidth for the attenuation L, where bandwidth is taken to mean the width in cps of the frequency range in which the actual attenuation is not less than L db. The table gives the proper relations for calculating the bandwidth for various values of $L_{max} - L$. The results are approximately correct if L is 20 db or more. If L is less than 20 db, the bandwidth is greater than the indicated value.

2. The Q is proportional to $(Af_o)^{1/2}$. For air under ordinary conditions, the Q is given approximately by Eq. (12.12).

$$Q \simeq 2.15\,(Af_o)^{1/2} \quad \text{(cgs units)} \qquad (12.12)$$

3. When the resonance pressure ratio is much greater than unity, this ratio is given approximately by Eq. (12.13a), which for air under ordinary conditions has the values shown in Eq. (12.13b):

$$r_o \simeq 1 + \frac{Ac\sqrt{\rho}}{8S\,(\pi\mu f_o)^{1/2}} \qquad (12.13a)$$

$$r_o \simeq 1 + \frac{6.0 \times 10^3\sqrt{A}}{S\,(f_o)^{1/2}} \quad \text{(cgs units)}. \qquad (12.13b)$$

4. The relation between the orifice area and the resonator volume, for air under ordinary conditions, is expressed by Eq. (12.14):

$$V \simeq 3.1 \times 10^7 \sqrt{A/f_o^2} \qquad \text{(cgs units).} \qquad (12.14)$$

TABLE 12.4

DATA FOR CALCULATING BANDWIDTHS OF RESONATOR ATTENUATION

L_{max} - L in db	Bandwidth for Attenuation not less than L
3	1.0 f_o/Q
6	1.7
10	2.6
12	3.9
15	5.5
18	7.9
21	11
25	18
30	32
35	56
40	100

In many cases the practical limitation on effective resonator design is set by the maximum allowable value for the cavity volume V. Therefore a possible design procedure, which is often satisfactory, is to assign the largest acceptable value to V, and then from Eq. (12.14) to find the approximate value of the corresponding orifice area A. The approximate values of Q and of maximum pressure ratio can be obtained then from Eqs. (12.12) and (12.13). If these approximate results indicate that the design is feasible, final, more accurate performance values may be obtained from the original equations.

Numerical Example. Suppose that attenuation is required in a frequency range centered about 30 cps, in a duct having a cross-sectional area S of 10^4 sq cm. It is possible to provide a resonator cavity having a volume as large as 1 cu meter (10^6 cu cm). The wall thickness is 1 cm. The system is to operate with air at 25°C, 760 mm pressure.

For initial calculations, the thickness of the orifice is neglected and the simplified equations, (12.12) to (12.14), are used. From these the following results are found:

Area of opening, $A = 842\ cm^2$
(radius 16.4 cm for circular opening)
Maximum pressure ratio, $r_o = 93$
Maximum attenuation, $L_o = 39.4$ db
$Q = 342$

The open area is of reasonable size, so that the design is thus far practical. It will be supposed that the attenuation appears to be sufficiently large to satisfy design requirements. Therefore more exact calculations are warranted. The area of 842 cm^2 will be retained and the other quantities will be found more precisely from Eqs. (12.3) to (12.9). In these calculations the orifice thickness will be taken into account. The results are the following:

$$R_A = 2.22 \times 10^{-5} \text{ cgs}$$

$$M = 4.0 \times 10^{-5} \text{ cgs}$$

$$V = 1.01 \times 10^6 \text{ cm}^3$$

$$Q = 340$$

$$r_o = 93.7$$

$$L_o = 39.4 \text{ db}$$

Since the orifice thickness is relatively small, these results agree closely with the original values in which t was neglected. The bandwidths for the various attenuations, as found with the help of Table 12.4, are shown below.

Attenuation db	Bandwidth in cps exceeds
36	0.09
30	0.23
21	0.7
14	1.6
9	2.9

These results are correct only when the input pressure, p_1, is sufficiently small to avoid nonlinear resistance effects at the orifice. These effects, which occur at large pressure levels, are discussed at the conclusion of this section.

Possible Errors in Resonator Calculations. The approximate relations given above for the calculation of resonator performance represent a simplification of a relatively complex subject. Accordingly discrepancies between calculated and observed results will sometimes be encountered. It has been found in one set of experiments, with resonators opening into a circular duct, that the resonance occurs at a frequency about 10% greater than the calculated value. In some configurations the observed value of the acoustic resistance R_A must be modified in the case of long tubes if the frequency is much less than a critical value which, in the case of a circular tube, is equal to 200/(radius in cm)^2. The latter condition is seldom encountered in the design of large resonators. When the particle velocity in the resonator is very large, nonlinearity occurs; that is, the effective value of the resistance R_A increases with increasing particle velocity. Some of these effects are discussed in the literature 6,7,8,9./

Another possible source of discrepancy between calculation and observation can be given by stating the principal assumptions which underlie the analysis of the attenuating action of a resonator attached to a duct. These assumptions are (1) The source end of the duct is so terminated that a signal traveling toward the source would experience negligible reflection; (2) The output end of the duct has a similarly low reflectivity;

(3) All dimensions of the resonator are much less than the wavelength of the sound; (4) The cavity is shaped so that a rapid change of cross-section occurs where the opening communicates with the cavity.

Nonlinearity is perhaps the most serious of the sources of error when the simple theory is applied to a resonator intended for noise control at high sound levels. No completely rigorous theory is available from which to compute the resistance under all conditions of nonlinear operation. When the particle velocity is not too great, however, the nonlinear contribution to the resistance is given approximately by the relation

$$R_{nonlinear} = \rho U/\sqrt{2}\, A^2 \qquad (12.15)$$

where U, the rms value of the volume current (cm^3/sec), may be computed from the equivalent circuit of Fig. 12.13. According to the equivalent circuit, the volume current is approximately $2p_1S/\rho c$ if the resonator impedance is small compared to the other impedances, as will be the case for a resonator which produces large attenuation. Therefore, the nonlinear resistance is given roughly by the expression

$$R_{nonlinear} = \sqrt{2}\, Sp_1/A^2c \qquad (12.16)$$

As before, p_1 is the rms sound pressure which would exist in the duct, at the frequency of resonance, in the absence of the resonator. The quantity $R_{nonlinear}$ should be added to R_A to obtain the total orifice resistance. Physically, $R_{nonlinear}$ represents a form of the Bernoulli pressure drop which is associated with the passage of fluid through an orifice or a constricted pipe.

It is useful to have a criterion for the value of the sound pressure at which nonlinear effects become important. For this purpose, let p_1' be the value of the pressure existing in the absence of the resonator, at the frequency of resonance, such that when the resonator is connected the total orifice resistance will be doubled as a result of nonlinearity. Under these conditions the values of the pressure ratio, r_o, and of the Q, will be approximately one-half of those given by the simple theory. Let p_o be the resonant frequency sound pressure which would be measured by a microphone placed in front of the orifice, after attachment of the resonator, under the same conditions. The two relations below, which give these pressure values, can be derived from Eq. (12.15) in conjunction with the equivalent circuit.

$$p_1' = \frac{\sqrt{2}\, A^2\, R_A\, c}{S} \left(1 + \frac{1}{r_o}\right) \qquad (12.17)$$

$$p_o = 2\sqrt{2}\ R_A^2\ A^2/\rho \qquad (12.18)$$

In these relations, R_A is the orifice resistance computed from Eqs. (12.6) without regard to nonlinearity.

When the values from the numerical example are used, it is found from Eqs. (12.17) and (12.18) that the critical pressure values for that example are

$$p_1' = 78 \text{ dynes/cm}^2 \quad (\text{SPL} = 112 \text{ db})$$

$$p_o = 0.84 \text{ dynes/cm}^2 \quad (\text{SPL} = 72.5 \text{ db})$$

There is no extensive experimental information on the performance of very large resonators under nonlinear conditions. It is therefore desirable to design with a large margin of safety when resonators must be used for noise control, or preferably to make preliminary measurements on a model which is as close as possible to full size.

12.7 Duct with Resonant Lining

High attenuation values can sometimes be obtained over a wider frequency range when a continuous duct lining with resonant properties is substituted for a single resonator. One possible arrangement for obtaining a resonant duct lining is illustrated in Fig. 12.14. The innermost portion of the lining is a perforated facing, which is characterized by its effective acoustic mass per unit surface area. Behind this facing (on the side away from the main air passage) is an acoustic blanket or tile which is characterized by a specific flow resistance. Finally there is an air space surrounding the resistive blanket or tile layer, which is characterized by the free volume per unit area of inner duct wall. These three elements form, in effect, a continuously distributed acoustic resonator. The optimum attenuation properties of the entire system can be realized only when the lining structure is broken by sound-isolating baffles as indicated in Fig. (12.14). The baffles

should be separated by a distance of not more than 1/10 of the wavelength of sound in free air, at the highest frequency at which attenuation is to be obtained. The cross-section may have any shape.

The specific acoustic resistance of the blanket will be denoted by R'. This quantity, which is the ratio of pressure drop to the particle velocity of air flow through the blanket, is ordinarily expressed in rayls (dyne sec/cm^3). The resistance value measured by steady air flow may be used. Some flow resistance data are given in Fig. 12.3.

The specific acoustic mass, which is the effective moving mass associated with unit area of lining, is given by the expression

$$M' = M_h/n \qquad (12.19)$$

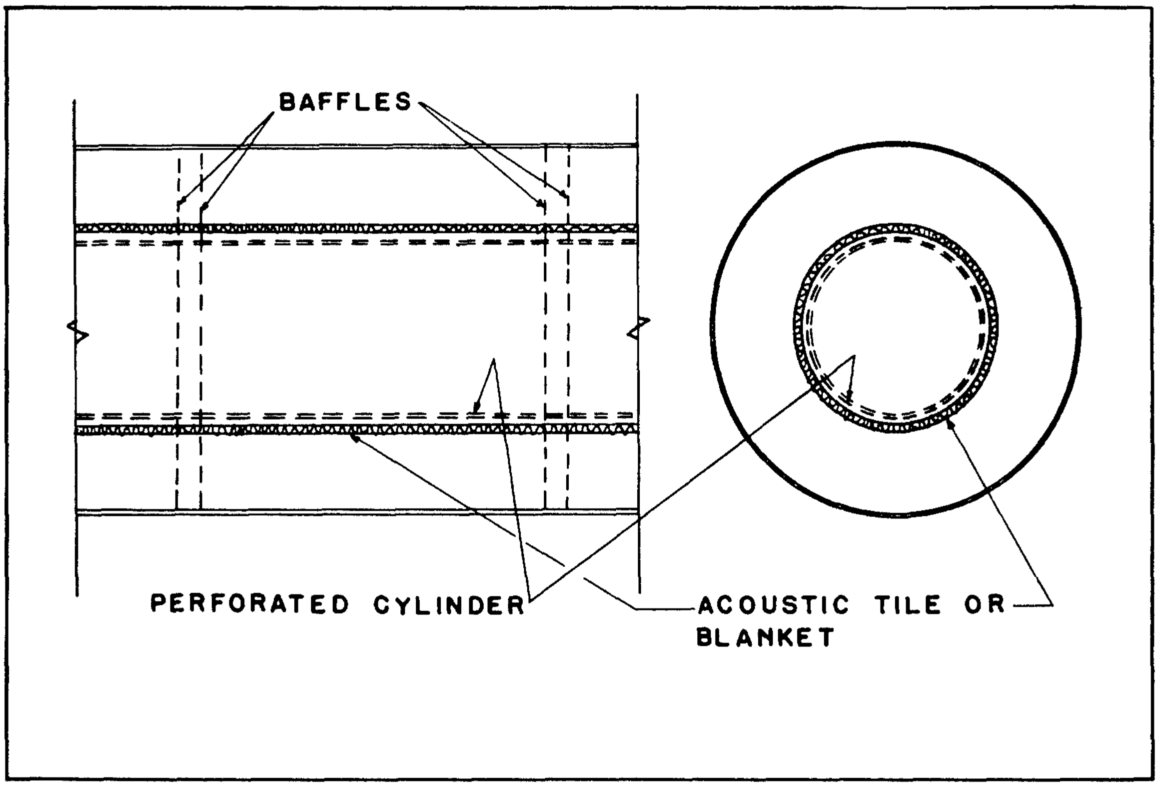
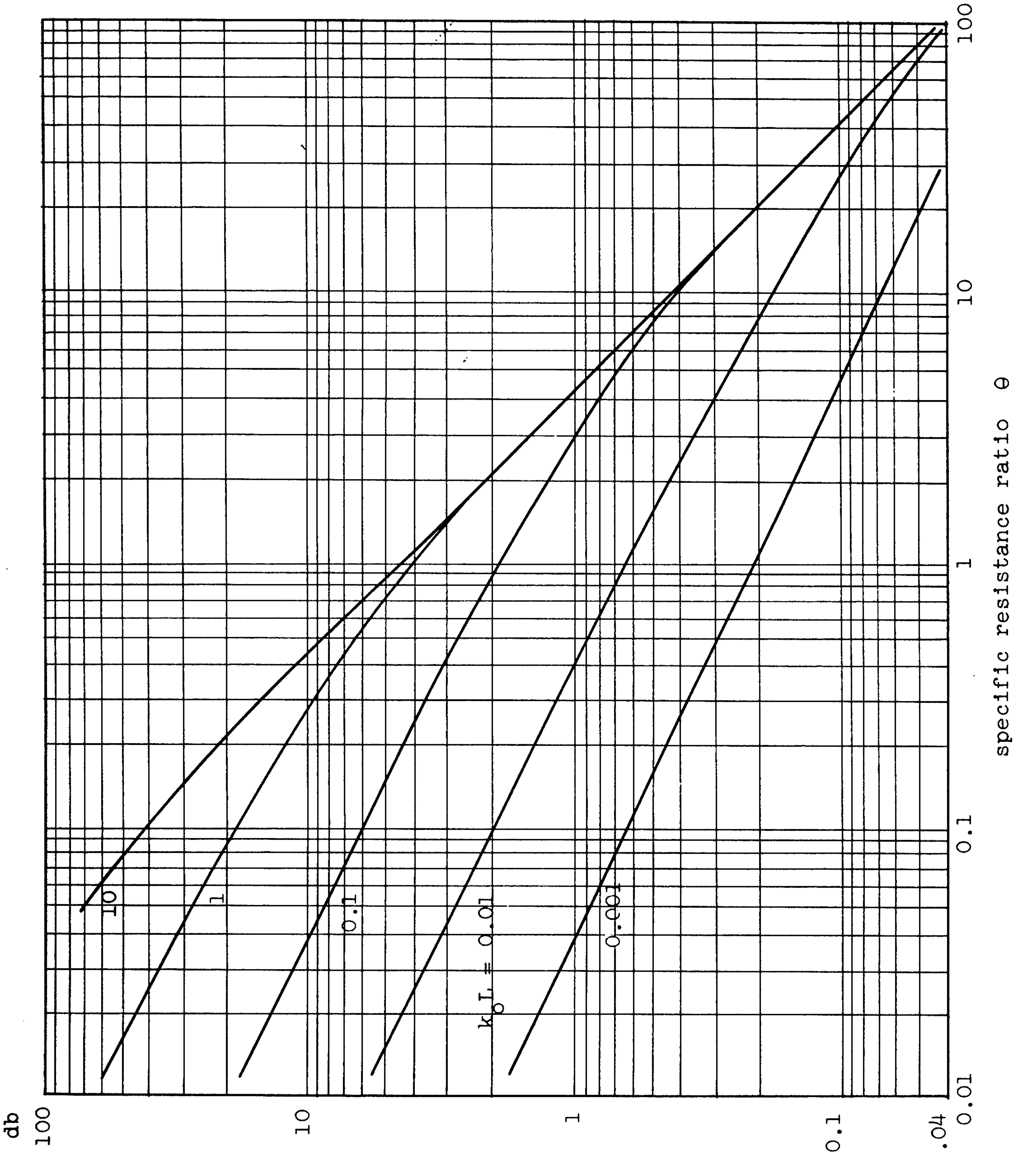


Figure 12.14

Sketch illustrating a duct with continuous resonant lining.


db
100
10
1
0.1
.04
0.01
0.1
1
10
100
specific resistance ratio θ
10
1
0.1
k_oL = 0.01
0.001

where M_h is the acoustic mass for one hole as computed from Eq. (12.4), and n is the number of holes per unit area of the facing. Equation (12.19) is valid only when the holes are separated by at least several times the diameter of one hole. In Eq. (12.19) it is assumed that the perforated facing is rigid. If the facing is flexible, the surface mass σ of the facing (in gm/cm^2) must be considered. In this case the specific acoustic mass is given by

$$M' = \frac{\sigma M_h/n}{\sigma + (M_h/n)} \tag{12.20}$$

A resonant lining can be formed with a flexible unperforated facing, for which the value of M' is σ .

The specific acoustic volume, which is the volume of the air backing space per unit area of the facing is denoted by V'. In the case where the air space is contained between concentric cylinders of diameters d_1 and d_2, where $d_2 > d_1$, the specific acoustic volume is

$$V' = (d_1/4) \quad [(d_2/d_1)^2 - 1] \tag{12.21}$$

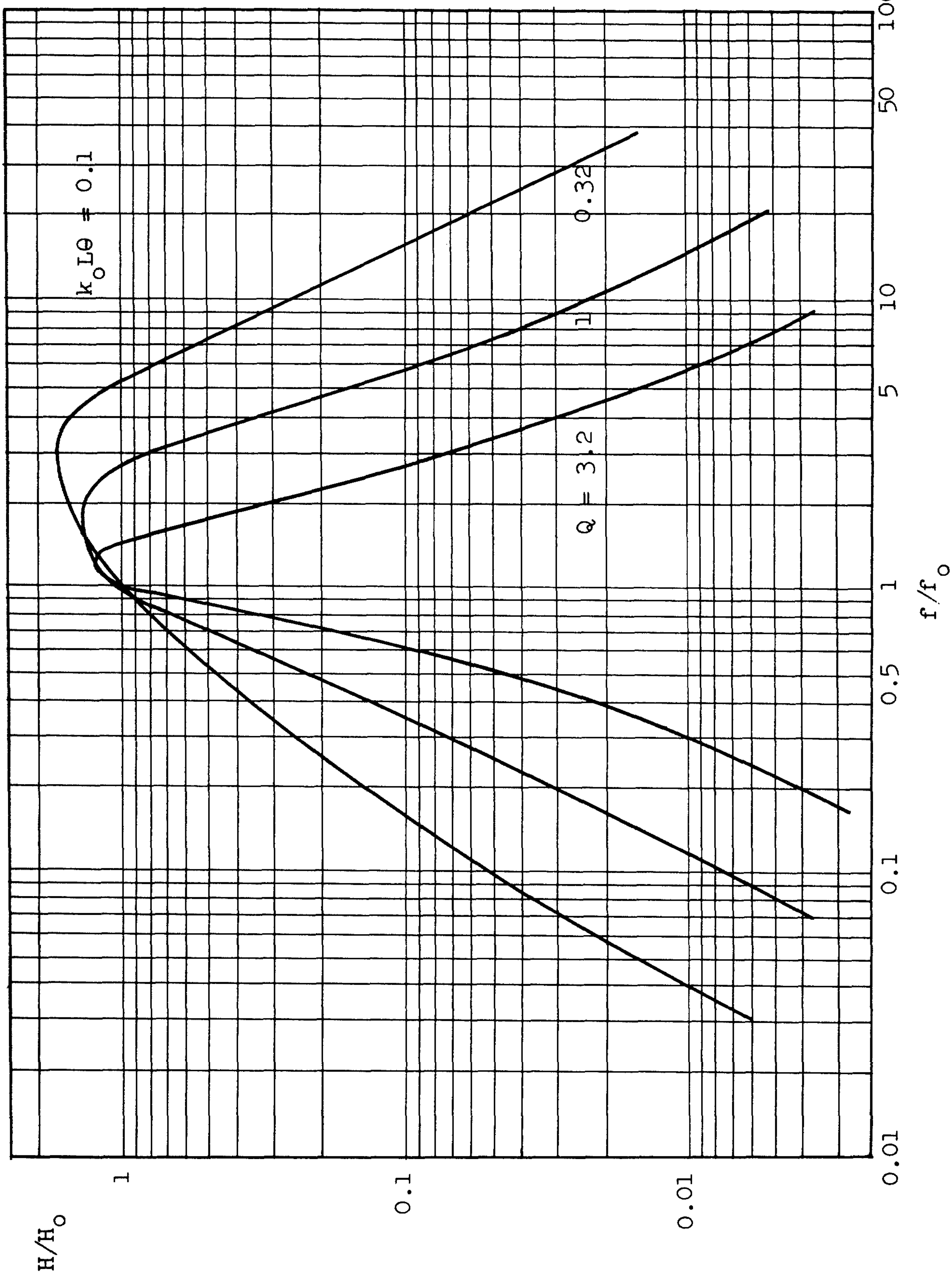
The specific acoustic compliance is

$$C' = V'/1.40 \quad P_o \tag{12.22}$$

The quantities R', M', and C' may be used in Eqs. (12.7) and (12.9) to obtain the frequency of resonance and the Q for the distributed resonator.

Figure 12.15

Resonance-frequency attenuation parameter H_o, in db, as a function of the specific resistance ratio Θ of a resonant duct lining. H_o is resonance-frequency attenuation in a length L of duct, where L = (Area in cm^2 of cross-section)/(Perimeter in cm). k_o is $2 \pi f_o/c$, where f_o is the frequency in cps for lining resonance and c is the speed of sound in open air, in cm/sec.


$k_oL\theta = 0.1$
0.32
1
Q = 3.2
f/f_o
H/H_o
0.01
0.1
0.5
1
5
10
50
100
1
0.1
0.01

The attenuation properties of the resonant-lined duct can be described by design charts in terms of the parameters defined below,

$$L = \frac{\text{cross-sectional area of air passage}}{\text{perimeter of air passage}}$$

$$k = 2\,\pi\, f/c = 2\,\pi/\lambda$$

where c and λ are the speed of sound and the wavelength respectively, in open air. When the frequency has the resonance value f_o, k becomes k_o. Also

Θ = $R'/\rho c$

H = attenuation in decibels for duct of length L

H_o = value of H at the resonance frequency f_o

The chart of Fig. 12.15 gives the value of the resonance-frequency attenuation factor H_o. The variation of the attenuation with frequency is shown by the charts of Fig. 12.16 (a,b,c).

The resonant lining is generally most useful when the specific resistance ratio Θ is made sufficiently large to reduce the Q to 3 or less, and thereby achieve attenuation over a broad frequency band. An attenuation of the order of 6 db per diameter unit of length is reasonable. Higher values can be obtained in a narrow frequency band by making Θ small. Commercially available porous blankets and tiles are found to have values of Θ of 0.2 or more.

The attenuating duct with continuous resonant lining should not be confused with the lined duct discussed in Sec. 12.2. Whereas the resonant lining is employed for the attenuation of sound having a wavelength many times the duct diameter, the lining described previously, which does not involve resonance effects,

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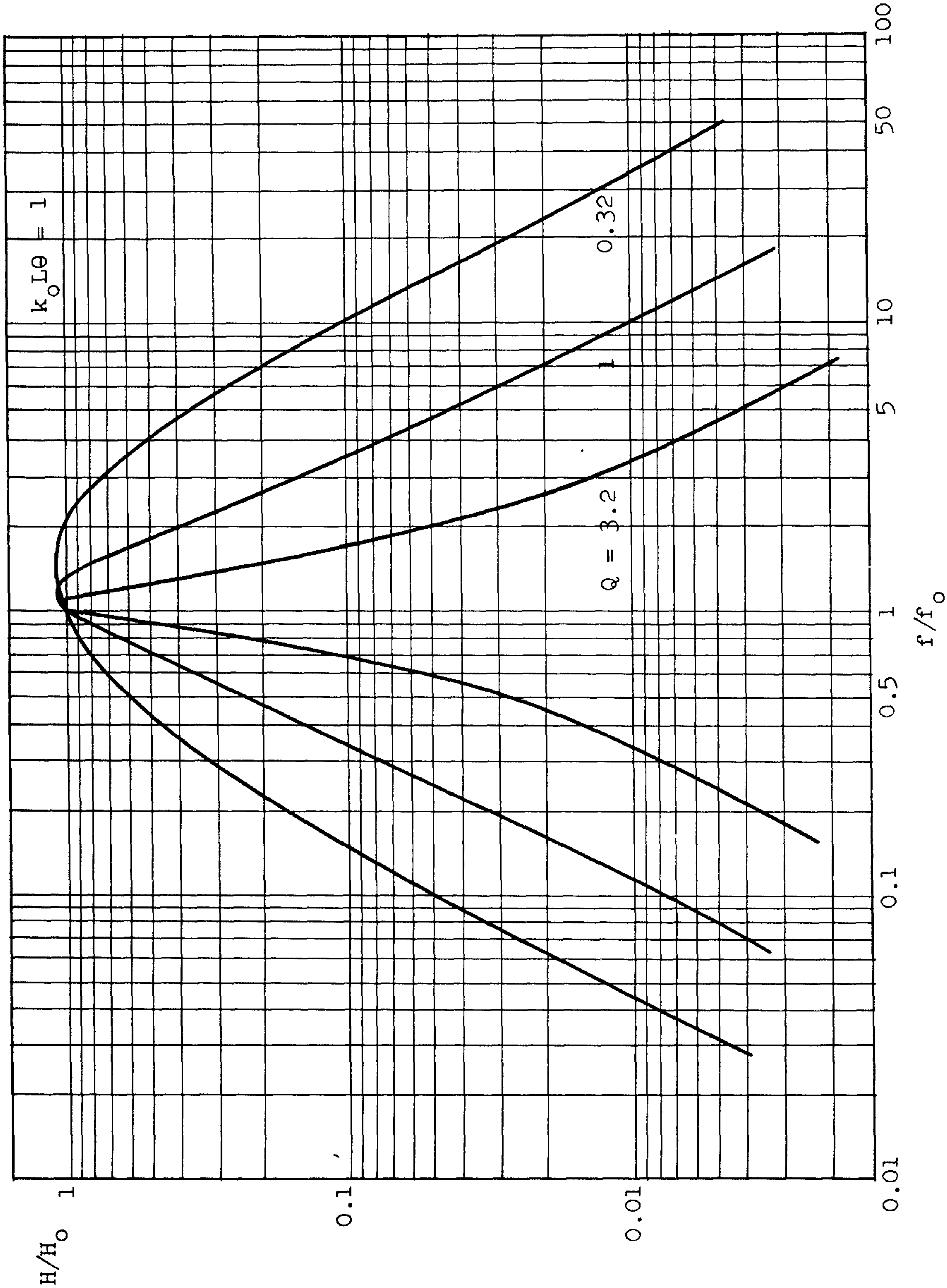
Figure 12.16 a

Relative attenuation values H/H_o, as a function of frequency)/(resonance frequency), for a duct with resonant lining. For a value of 0.1 of the parameter $k_oL\Theta$ and various values of Q.


H/H_o
1
0.1
0.01
0.01
0.1
0.5
1
5
10
50
100
f/f_o
k_oLθ = 1
Q = 3.2
1
0.32

produces a large attenuation when the wavelength is between one and two times the duct width. The maximum of attenuation with respect to frequency in the latter case is the result of a complicated phenomenon associated with distortion of the wavefronts in the duct.

Example of Resonant Continuous Duct Lining. Suppose that a duct is to be designed to produce sound attenuation in a band of frequencies centered around 150 cps, subject to the following requirements:

Operating pressure and temperature, 760 mm and 25°C.

The air passage, or duct proper, shall be circular in section, with a diameter of 20 cm.

The outer shell shall be circular in section, with an inside diameter of 40 cm.

An attenuation of at least 0.2 db/cm is required in the widest possible frequency band.

One estimates, by inspection of Fig. 12.16 (a,b,c) that f_o should be made 100 cps in order to center the attenuation band at 150 cps. Also it is estimated that the maximum attenuation should be 0.3 db/cm in order that the required value of 0.2 db/cm shall exist over a wide frequency band. Then the following values are found for the parameters:

$$L = \frac{\text{Area}}{\text{Perimeter}} = 5 \text{ cm}$$

$$k_o = 0.0185 \text{ cm}^{-1}$$

$$k_oL = 0.0925$$

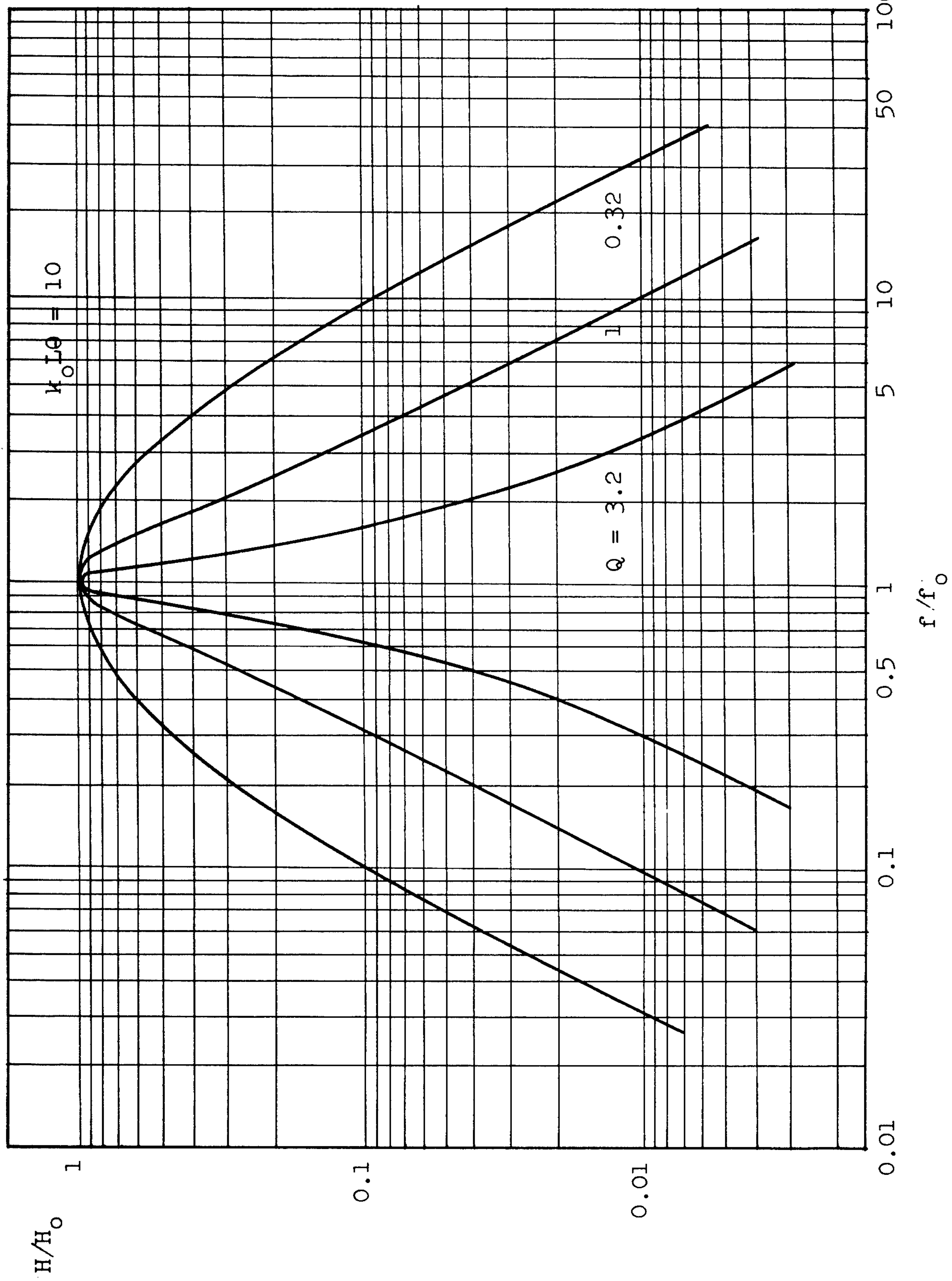
$$H_o = 0.3 \text{ db/cm} \times L = 1.5 \text{ db.}$$

The specific acoustic volume is (from Eq. (12.21),

$$V' = 15 \text{ cm.}$$

Figure 12.16 b

Relative attenuation values H/H_o, as a function of (frequency)/(resonance frequency), for a duct with resonant lining. For a value of 1.0 of the parameter $k_oL\Theta$. Various values of Q.


k_oLθ = 10
0.32
1
Q = 3.2
0.01
0.1
0.5
1
5
10
50
100
f/f_o
1
0.1
0.01
H/H_o

According to Eq. (12.22), the specific acoustic compliance is

$$C' = 1.05 \times 10^{-5} \qquad \text{(cgs units)}$$

The specific acoustic mass value which will place the resonance at 100 cps is, according to Eq. (12.7),

$$M' = 0.240$$

If there is any doubt that the required values of M' can be obtained with a practical perforated facing, the designer must at this point investigate available materials having the specific acoustic mass indicated.

According to Fig. 12.16 (a,b,c), the required value of H_o will be obtained with Θ equal to 1.4. Q is now calculated from Eq. (12.9) which gives a value of Q = 2.6. The value of $k_oL\Theta$ is 0.13. Therefore, the chart of Fig. 12.16a will approximately describe the frequency dependence of the attenuation. According to this chart, (H/H_O) exceeds 0.67 for (f/f_o) between 0.9 and 2.0, when the Q is 2.6. Therefore, the attenuation exceeds 0.2 db/cm for frequencies between 90 and 200 cps.

The design of the perforated facing is next investigated. By combining Eqs. (12.4) and (12.19), it is found that the specific acoustic mass for a perforated rigid facing is given by

$$M = \frac{e}{n}\left[\frac{t + 0.96\sqrt{A}}{A}\right] \qquad (12.23)$$

where A is the open area of one hole. It will be assumed that the lining is made of material 0.1 cm thick (with adequate stiffening). Combinations of values of A and n (number of holes per cm^2), which give the required value of M', are found from Eq. (12.23). For example, in the present case the necessary value for M' could be obtained with the following two combinations:

Figure 12.16 c

Relative attenuation values H/H_O, as a function of frequency)/(resonance frequency), for a duct with resonant lining. For a value of 10 of the parameter $k_oL\Theta$ and various values of Q.

	Combination 1	Combination 2
n, number holes/cm^2	1	0.1
A, hole area, cm^2	6.10×10^{-4}	9.50×10^{-3}
Hole spacing on square lattice, in.	0.4	1.25
Hole diameter, in.	0.011	0.043

To secure a workable final design, it may be necessary to modify the values slightly in order to make use of the available perforated facings and acoustic blankets which correspond most nearly to the desired specifications.

12.8 Sound Propagation in the Atmosphere

Noise control methods considered in previous sections have concerned sound fields which were contained within artificial boundaries, i.e., sound in ducts, resonant chambers, etc. In this section sound propagation in an extended medium will be discussed. Sometimes the simplest and cheapest method of noise control is to increase the distance of sound travel from the source to the receiver.

Attenuation by distance is the result of two basic phenomena: inverse-square spreading and sound absorption. The first is only an apparent attenuation; the same amount of acoustic energy is present, but it is spread over a greater area, as the distance from the source increases. The second is a true energy loss resulting in the conversion of acoustic energy to heat energy. Close to an actual noise source the situation is usually far more complicated than the simple inverse-square picture since most noise sources have an appreciable size. "Near-field" sound levels may be much higher than predicted on the basis of simple inverse-square attenuation (see Chapter 3).

At a large distance from its source, a free-field sound wave in a homogeneous and undisturbed medium will continuously diverge so that the sound intensity will fall off as the inverse

square of the distance from the source as explained in Chapter 3. The sound pressure, being related to the square root of the intensity, will fall off inversely with the first power of the distance, and hence will be a decrease of 6 db with each doubling of the distance from the source.

Absorption losses are due chiefly to viscosity, heat conduction, and molecular energy abosrption. The absorption process is a function of frequency, relative humidity, temperature, and the composition of the atmosphere. For example, at 10,000 cps the attenuation due to absorption in dry air on a moderately warm day can be approximately 9 db/100 ft. At higher temperatures this attenuation may be even larger. Several tables have been constructed for the evaluation of this attenuation. 8/ To a first approximation the attenuation due to heat conduction and viscosity is given by

$$\alpha_c \simeq \frac{3.6 \times 10^{-7}(180 + T)}{\lambda^2} \quad \text{db/ft} \qquad (12.24)$$

where λ is the wavelength in feet and T is the temperature in degrees centigrade.

There will be additional effects on the propagation of airborne sound 9,10,11/ due to dynamic meteorological conditions. Wind and temperature variations are of principal importance, for they tend to bend the sound waves from their original paths, with a consequent distortion of the normal radiation pattern of a source. In moving air the velocity vector of a sound wave with respect to a fixed point on earth is equal to the vector sum of the velocity of sound in still air and the velocity of the moving air. If the wind velocity increases with height (positive vertical wind gradient), sound rays traveling with the wind will be bent downward, increasing the sound level observed on the ground at a point remote from the source. Conversely, rays that travel against the wind will be bent upward with a consequent reduction in the sound level observed.

Vertical temperature gradients can result in an effective bending of sound waves that is similar to that produced by wind, since the speed of sound is directly proportional to the square root of the absolute temperature of the air. For a negative temperature gradient, i.e., a decrease of temperature with height, sound rays radiating from a source on the ground will be refracted upward. If a temperature inversion occurs, such as is commonly found in areas where early morning or late evening

fogs are prevalent, an appreciable amount of sound energy will be refracted downward by the warmer upper air layers. It can be shown that the sound intensity under such circumstances will vary as the reciprocal of the square root of the distance rather than as the reciprocal of the distance as discussed previously. This condition is responsible for the long distances over which sound is propagated in certain northern regions.

Both wind and temperature gradients will be found to occur simultaneously, to vary with the time of day, and to be mutually dependent. In some situations these effects result in rather anomalous wave propagation phenomena.

The outdoor propagation of sound becomes an even more complicated problem when effects due to the surface of the earth are considered in addition to wind velocity and air temperature. Some theoretical studies of the effect of the surface impedance upon wave propagation have been made 12,13/ and laboratory data have been obtained for some of the phenomena. 14/ Calculations have been published for the attenuation of sound traveling over snow-covered ground. 15/ An experimental study of the propagation of sound in a jungle has been carried out. 16/ The latter two studies report attenuation significantly greater than that experienced in the propagation of sound over unobstructed hard earth.

12.9 The Isolation Wall

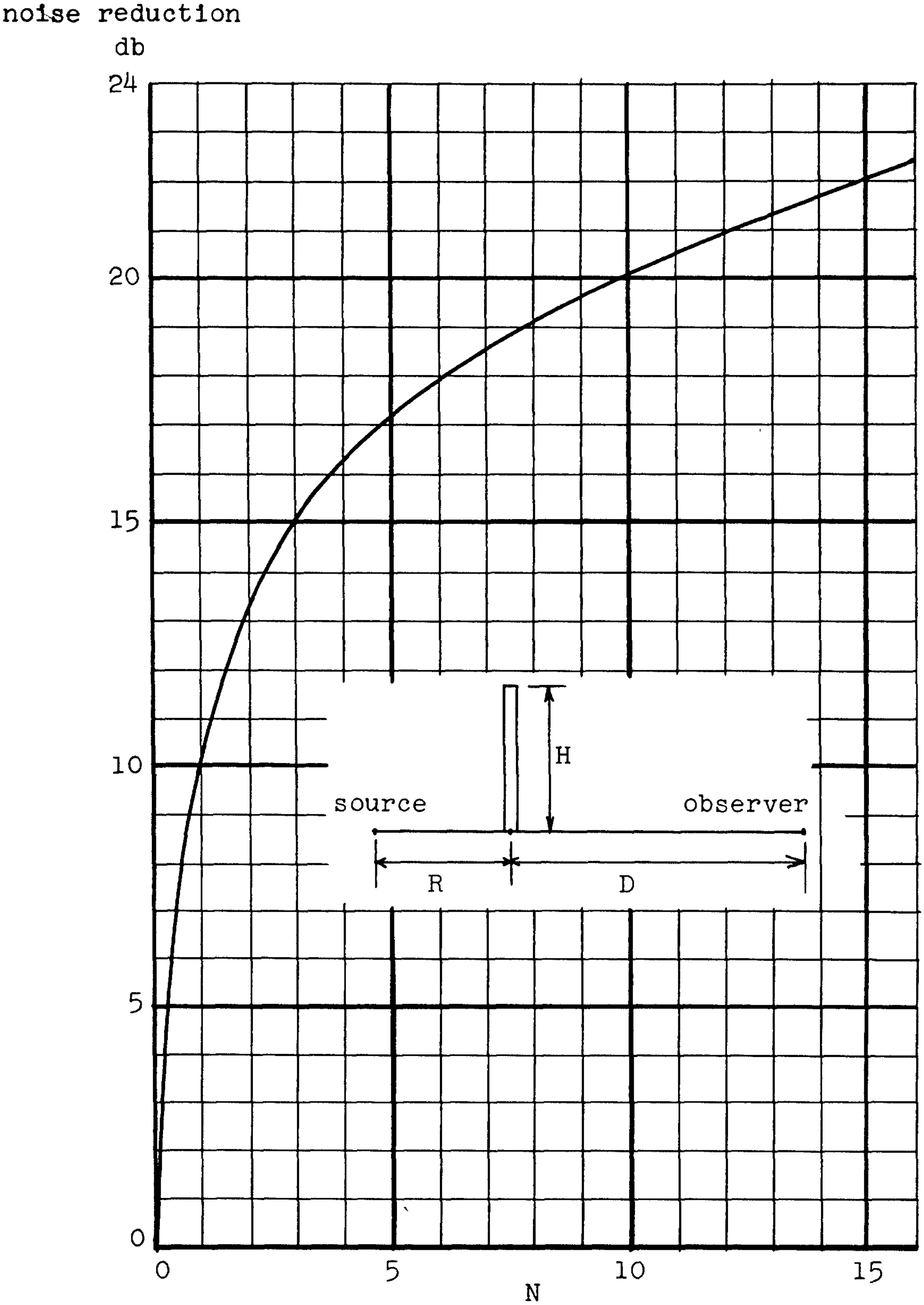
A wall is an effective outdoor sound barrier when its height is suitably related to the wavelength and to the distances involved. A design chart for isolation walls has been published 17/ and is reproduced in Fig. 12.17. The predictions of the chart have been verified experimentally. The values given below, calculated with the help of the chart, indicate the performance which may be obtained.

wall height, h = 15 ft

distance of source to wall, R = 25 ft

distance of observer to wall, d = 100 ft or more.

Frequency, cps	100	300	1000	3000	10,000
Loss, db	9	14	18	>25	>25



$$N = \frac{2}{\lambda}\left[R\left(\sqrt{1+H^2/R^2}-1\right)+D\left(\sqrt{1+H^2/D^2}-1\right)\right]$$

$$\approx \frac{H^2}{\lambda R} \quad \text{IF} \quad D \gg R \geq H$$

If the observer is more than a few hundred feet from the wall, the sound level may be affected strongly by the wind and temperature effects described in Sec. 12.8.

12.10 Directionality

At frequencies such that the wavelength of sound is smaller than the greatest dimension of the effective radiating surface of a source, the distribution of sound energy is not uniform in all directions. This tendency to directionality at higher frequencies can often be used effectively in noise control.

Adequate theories are available to describe the directional distribution of sound intensity when the source is a plane wave in a flanged, circular 18,19/ or rectangular pipe, or a plane wave in an unflanged circular pipe. 20/

Noise control problems may involve sound radiated from a large opening in a building or from the open end of a large air duct. Openings 30 ft square and larger are commonly encountered. The theories cited above may be expected to apply for plane waves of sound of wavelength no smaller than the perimeter of the opening, provided that there is no air flow in the duct and no temperature gradient at the open end. Even if there are no complications from air flow or from the presence of hot gases, there is danger in applying the theories to the radiation of high-frequency noise from a large opening, because of the plane-wave assumption in the theories. The plane-wave condition will not exist when sound coming from within a room or a duct is radiated through an opening much larger than the wavelength, but instead multiple reflections within the structure will produce a distribution of sound pressure which may exhibit large irregularities in phase and amplitude. Consequently, it is usually necessary to rely instead upon the experience of many measurements in order to evaluate the probable effect of directionality in a given situation.

Figure 12.17

Noise reduction in db due to erecting a wall of height H above the line of sight between sound source and observer (19). The chart is plotted in terms of the parameter N, which is defined on the chart.

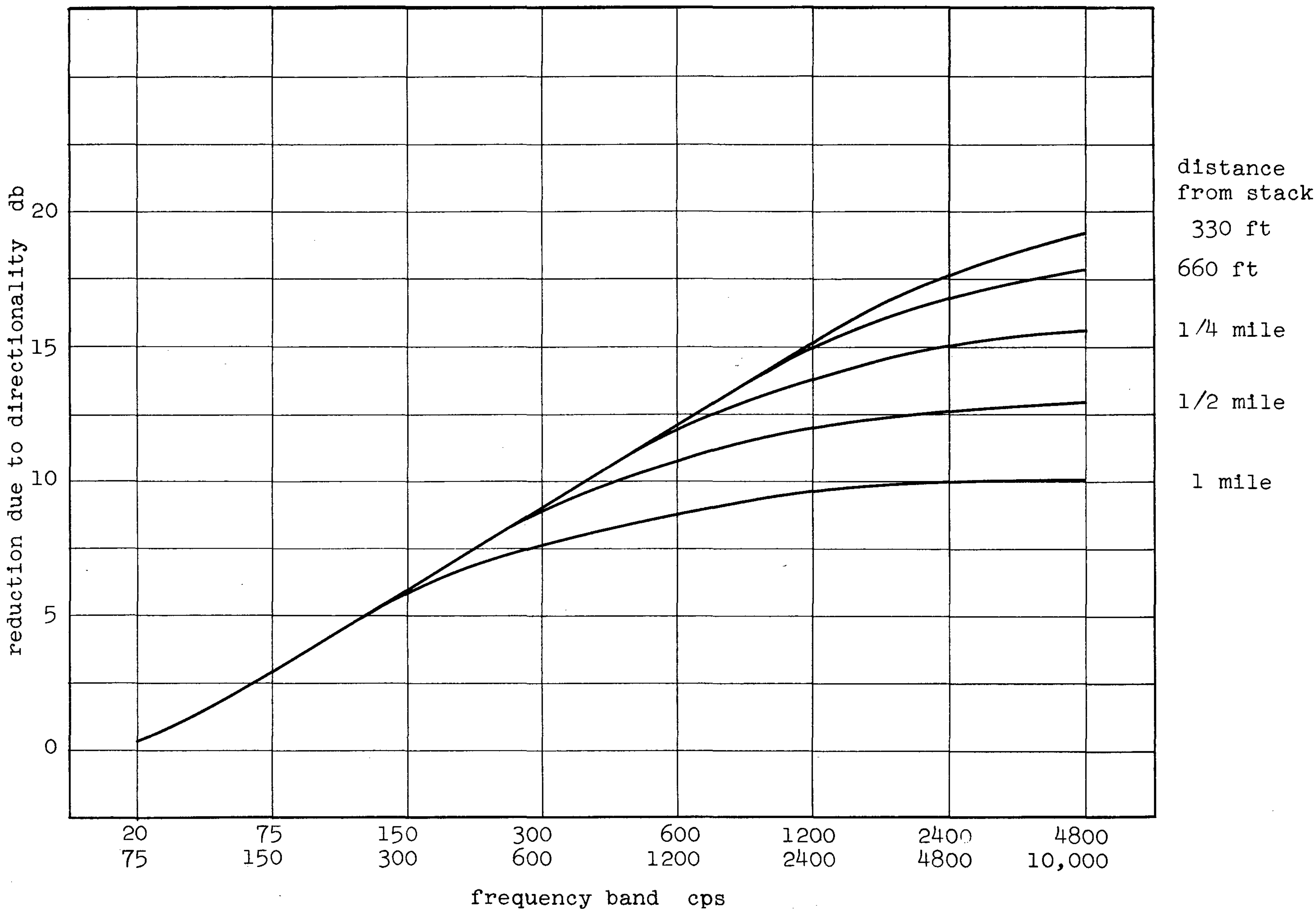
Measurements indicate that the greatest intensity exists in the forward direction, and most of the sound power is transmitted within 90° of the forward direction. The relative concentration of power in the forward direction depends upon the frequency, and increases rapidly with increasing frequency. Air flow and hot gases influence the relative concentration, however, and the general effect of both is to reduce the forward concentration. Hot gases cause the sound waves to be refracted away from the forward direction, and therefore the effective reduction of the forward concentration of sound will depend upon the distance of the observer from the opening.

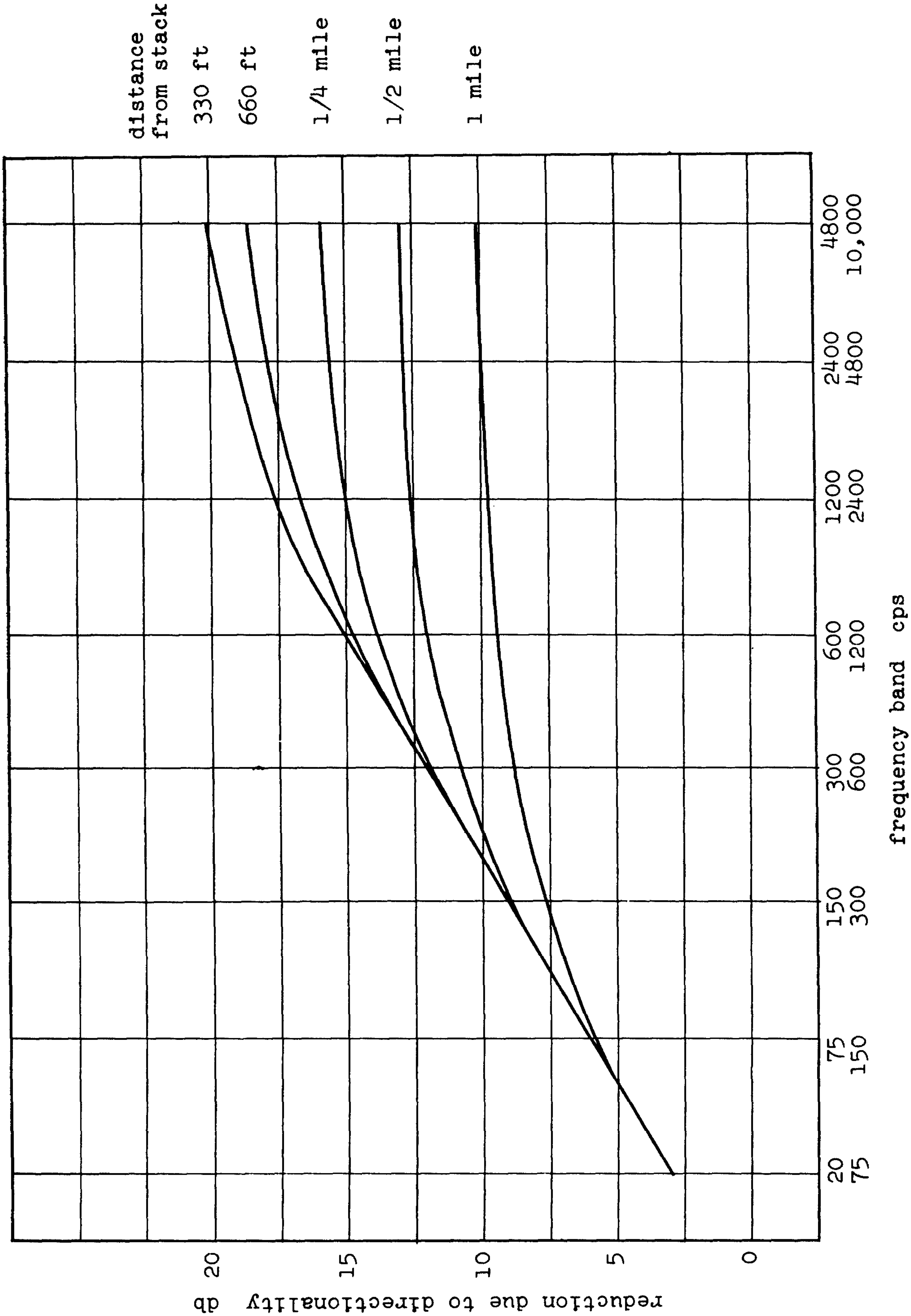
The effects of directionality at large distances are further modified by atmospheric refraction, due to temperature and wind-velocity gradients. Atmospheric refraction may greatly increase the intensity of the sound received at ground level, at large distances from a vertically directed stack. When detailed information on atmospheric refraction is not available, the acoustical engineer must make an arbitrary allowance for this effect by assigning smaller design values to the directionality expected at large distances.

Design charts, based upon engineering experience with the directional radiation of noise from openings in ducts or enclosures, is summarized in Fig. 12.18 (a,b,c). These figures show, on an octave-band frequency scale, the difference in db between the SPL which is actually observed in the side direction (90°) and the SPL which would be observed if the total power of the source were radiated uniformly in all directions. This difference is the directivity index for 90°. The results can be applied for noise radiated from a vertical stack. The information as given includes the effects of both stack perimeter and distance from the stack, and is valid for stacks with air flow. The directivity reduction of SPL with hot gases (exhaust stack) is 5 db less than the reduction with air at normal temperatures (intake stack).

Figure 12.18 (a)

Estimate of the noise reduction at ground level resulting from directional radiation of sound from a vertical exhaust stack containing hot gases. For an approximately square or circular opening of perimeter 25 ft. Various distances from the stack. Add 5 db to obtain values for vertical intake stacks containing normal gases.


reduction due to directionality db
20
15
10
5
0
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps
distance
from stack
330 ft
660 ft
1/4 mile
1/2 mile
1 mile


distance from stack
330 ft
660 ft
1/4 mile
1/2 mile
1 mile
reduction due to directionality db
0
5
10
15
20
frequency band cps
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000

An illustration of the use of the charts is given in the numerical example of Sec. 12.14.

12.11 Combined Treatments

Actual problems in control of airborne sound are usually solved by an appropriate combination of several of the control measures described in the preceding sections. The greatest economy in sound treatment is usually obtained when some portion of the total attenuation required between source and observer is provided by the various outdoor effects which have been described (inverse-square attenuation, directionality, surface attenuation, wind and temperature gradients). The cost aspects of the final compromise (between outdoor attenuation and attenuation by sound control treatments) are described briefly in Chapter 9.

That part of the total attenuation which is produced by acoustical treatments is in turn almost always obtained by a combination of several elements. Each of the elements is effective over only a portion of the entire range of audio frequencies. For example, in an extreme case, a single system may include in sequence a duct section with several resonators, smaller ducts formed by partitions and lined with absorbent material, sets of parallel baffles, and one or more lined bends. The treatment required for an aircraft engine test cell often includes a lined bend, sections of lined duct, and one or more sets of parallel baffles. An illustrative numerical example of a composite treatment for an engine test cell is given in Sec. 12.14.

It is possible to design special acoustical treatments, for air passages, in which attenuation elements operating on several different principles are combined, rather than placed sequentially in separate lengths of duct. A compound treatment

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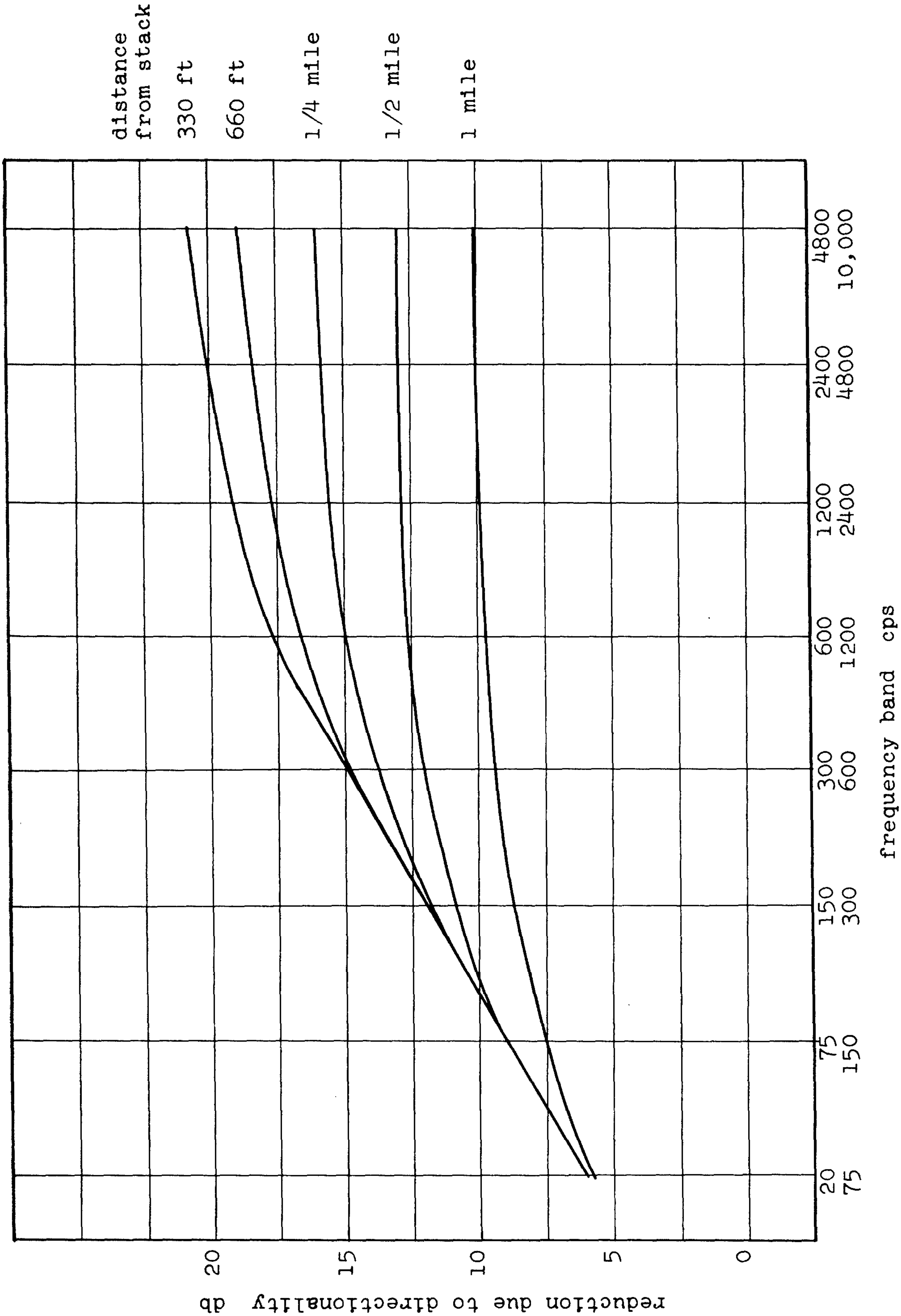
Figure 12.18 (b)

Estimate of the noise reduction at ground level resulting from directional radiation of sound from a vertical exhaust stack containing hot gases. For an approximately square or circular opening of perimeter 50 ft. Various distances from the stack. Add 5 db to obtain values for vertical intake stacks containing normal gases.

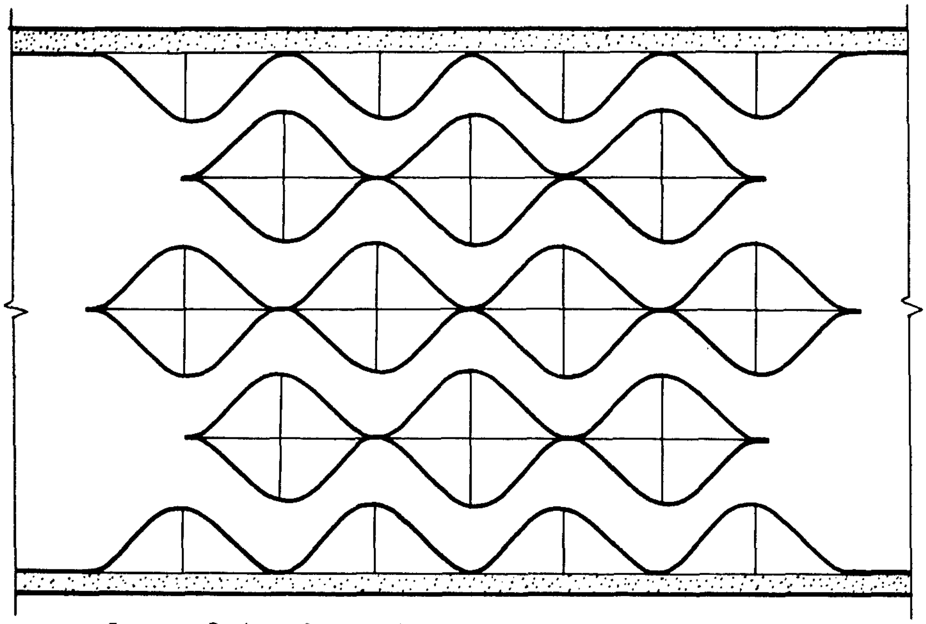

distance from stack
330 ft
660 ft
1/4 mile
1/2 mile
1 mile
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
frequency band cps
0
5
10
15
20
reduction due to directionality db

of this kind may in some instances be less expensive than a sequential treatment. An example of a compound attenuator is a sound-control system known by the proprietary name "Soundstream absorber". The structure, illustrated in Fig. 12.19, is intended for installation in a rectangular air duct. The longitudinal partitions, which are parallel to one pair of walls of the containing duct, effectively subdivide the main passage into smaller ducts of long, narrow cross-section. The open area is 42 percent when the partition spacing is 4 ft 10 in., which is the nominal design value. The attenuation produced by Soundstream absorber is attributable to a combination of four mechanisms, equivalent to a series of lined bends, a lined (tuned) duct, a periodic structure 21/, and volume resonators. These are discussed separately in Secs. 12.2, 12.4, 12.6 and 12.7.

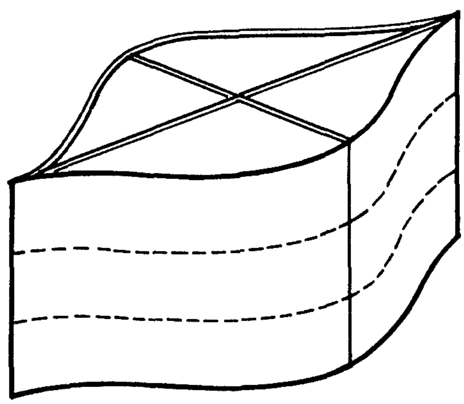
Measurements taken on scale models of Soundstream absorber under laboratory conditions are shown in Fig. 12.20 for the equivalent of 16, 20, and 24 ft lengths. The observed total attenuation agrees closely with estimated theoretical values. Preliminary measurements on a full-scale industrial installation of Soundstream absorber are shown in Fig. 12.21. The measurements were taken using a wide-band noise source located at one end of the Soundstream absorber treatment and a microphone and octave band analyzer at the other end. At the microphone position, the SPL was not constant across the width of the duct. This is indicated by the vertical bars in Fig. 12.21. The line which is drawn is an estimate of the average attenuation achieved. It is believed that the full attenuation predicted by the model tests can be achieved in installations where close attention is given to vibration isolation between adjacent units and to selection of materials.

Figure 12.18 (c)

Estimate of the noise reduction at ground level resulting from directional radiation of sound from a vertical exhaust stack containing hot gases. For an approximately square or circular opening of perimeter 100 ft. Various distances from the stack. Add 5 db to obtain values for vertical intake stacks containing normal gases.



plan of typical test cell installation



four Soundstream absorber units

12.12 Effects of Ambient Temperature and Pressure on Attenuation

Since the acoustical properties of air depend upon the temperature and the average pressure (usually the barometric pressure), it follows that the performance of components designed for attenuation of airborne sound is affected by these quantities. The dependence upon temperature and pressure of several of these acoustical properties of air is indicated by the relationships below. The absolute temperature (Kelvin scale) is indicated by K; the barometric pressure in mm of mercury is represented by P_b; the temperature on the centigrade scale is designated T.

Speed of sound: $c = 3.3145 \times 10^4 \sqrt{\frac{K}{273}}$ cm/sec

Density: $\rho = 0.001293 \left(\frac{P}{760}\right)\left(\frac{273}{K}\right)$ gm/cm3

Characteristic impedance: $\rho c = 42.86 \sqrt{\frac{273}{K}} \left(\frac{P_b}{760}\right)$ rayls

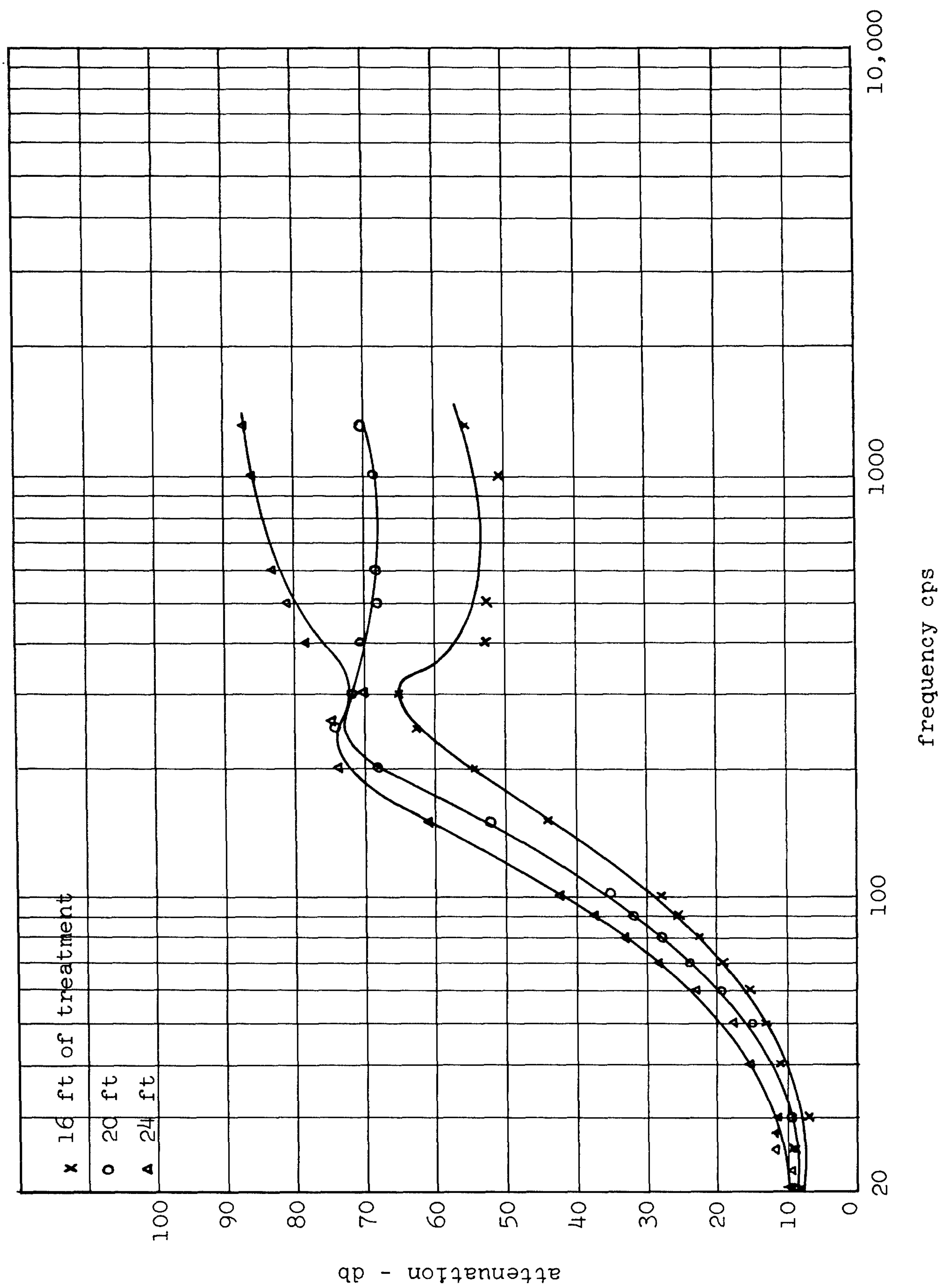
Viscosity: $\mu = 1.819 \times 10^{-4} + 0.00494 \times 10^{-7} (T - 20)$ gm/cm sec

The experimental data for the viscosity of air do not correspond accurately to a proportional dependence on Kelvin temperature. For purposes of discussing temperature effects in attenuators in general terms, however, it will be assumed that the viscosity is proportional to K. According to the simple kinetic theory of gases, μ is proportional to $\sqrt{K}$. The specific acoustic resistance of an acoustical blanket is proportional to the viscosity of air, and the viscosity appears in the design equations for resonators.

The effects of temperature and pressure on the performance of a number of the attenuating systems can be found by theoretical analysis. The results are given in Table 12.5, which shows how the important quantities of several systems are proportional to K and P_b. The symbols r_o and Q have the same significance as in Eqs. (12.3) and (12.9). In the case of systems having a resonance, the symbol f_o denotes the frequency of resonance; for lined ducts and for parallel baffles, f_o denotes the frequency of maximum attenuation; for the duct bend, f_o denotes the frequency at which the attenuation begins to rise rapidly.

Figure 12.19

Sketch of Soundstream absorber structure.



× 16 ft of treatment
o 20 ft
▲ 24 ft
attenuation - db
100
90
80
70
60
50
40
30
20
10
0
frequency cps
20
100
1000
10,000

TABLE 12.5

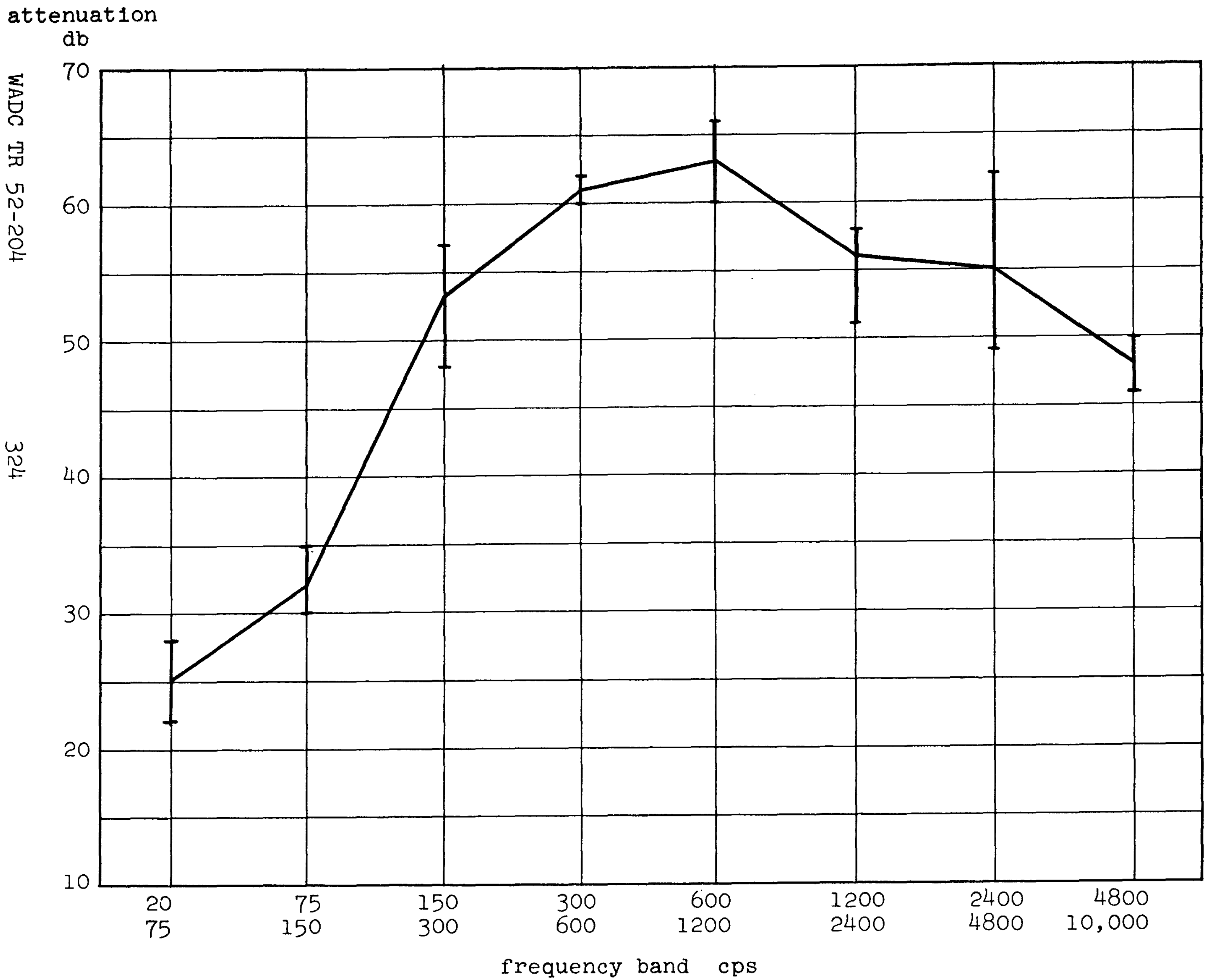
EFFECTS OF AMBIENT PRESSURE AND TEMPERATURE UPON PERFORMANCE OF STRUCTURES FOR ATTENUATING AIRBORNE SOUND

STRUCTURE	FUNCTIONAL RELATIONSHIPS
Lined ducts or Parallel baffles	$f_o \propto \sqrt{K}$ db loss $\propto$ $1/K$ *
Duct bend	$f_o \propto \sqrt{K}$ (Characteristic curve slides along the frequency scale but amount of loss is not seriously affected)
Duct with attached resonator	$f_o \propto \sqrt{K}$ $r_o \propto \sqrt[4]{P_b^2/K^3}$ $Q \propto \sqrt[4]{P_b^2/K^3}$
Duct with resonant lining (resistance furnished by blanket or tile)	$f_o \propto \sqrt{K}$ $Q \propto P_b\sqrt{1/K^3}$ Db loss $\propto \sqrt[4]{P_b^2/K^3}$ if $\theta k_o L \ll 1$ Db loss $\propto P_b\sqrt{1/K^3}$ if $\theta k_o L \gg 1$

* Tentative; no complete analysis available. A range extending from $1/\sqrt{K}$ to $1/\sqrt{K^3}$ has been observed.

Figure 12.20

Attenuation in db of several different lengths of Soundstream absorber assembled with 4 ft 10 in. on-center spacing; measured under laboratory conditions at a temperature of 70^oF.



attenuation
db
70
60
50
40
30
20
10
20
75
75
150
150
300
300
600
600
1200
1200
2400
2400
4800
4800
10,000
frequency band cps

Analysis of the effects is highly complicated in the case of lined duct or of parallel baffles, and no accurate general expressions have been obtained. A change of temperature in one of these structures not only shifts the frequency of maximum attenuation, but alters the shape of the attenuation vs. frequency curve. The data for the attenuation by lined ducts and parallel baffles, given in Secs. 12.2 and 12.3, are approximately correct for temperatures near 25°C. A temperature correction should be applied to these data when designing systems for operation at elevated temperatures. No data are available regarding the exact change in shape of the attenuation curve.

12.13 Precautions to be Observed in the Installation of Attenuating Systems

The installation of the various structures for sound attenuation in air passages must be planned and executed so as to avoid acoustic "leaks" or "flanking transmission", and to minimize deterioration in use. A list of design features which must be considered is given below.

Features tending to increase flanking transmission:

- Air leaks in enclosure or duct,
- Insufficient transmission loss of walls of enclosure or duct,
- Lack of vibrations breaks in duct walls,
- "Telegraphing" of sound by longitudinal metal beams, plates, or pipes within the attenuator,
- Failure to extend parallel baffle treatment to duct walls.

Features tending to increase deterioration in use:

- Acoustical material unsuited for continued operation at prevailing temperature,
- Acoustical material inadequately protected against wind erosion.
- Insufficient provision for preventing packing of acoustical material of wool type,

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Figure 12.21

Attenuation in octave bands of an industrial Soundstream absorber installation.

Acoustical material subject to chemical action or to irremediable fouling,

Acoustical material and containing structures inadequately protected against vibration.

Some tentative general recommendations in connection with these problems are given in the following sections.

Air Leaks. The power transmitted by an opening, for a given incident intensity, is roughly proportional to the area of the opening. This is only approximate when the width of the opening is much less than a half wavelength. The importance of sealing air leaks can be shown by consideration of the fact that the power transmitted by an open area one sq in. is nearly equal to the power transmitted through an open area of 1000 sq ft to which is attached a 50 db attenuator, assuming the incident sound intensity being the same in both cases.

Transmission Through Walls. The maximum possible reduction in total radiated power is obtained with an attenuating duct only when the power radiated via other paths (through the enclosure walls, through the duct walls, and from other openings) is negligible in comparison to the power transmitted through the duct. This does not mean that in practice all radiation must come through a single acoustic path. There is in general a most economical design compromise which allows some power to be transmitted by each of several paths (Chapter 10). Some requirements upon the minimum permissible transmission loss of the walls must be imposed in every case and failure to meet these can render an attenuating duct largely ineffective. Usually the power transmitted via all other paths should not be greater by more than 5 db than the power transmitted through the duct in question.

Flanking Transmission by Walls. Another aspect of wall transmission is "flanking transmission". This mechanism is illustrated by the passage of vibration along a duct wall. This vibration, produced at the source end where the interior sound level is high, is generally transmitted with little attenuation along the duct wall unless vibration breaks are inserted. Thus the intensity of the sound radiated at the low-energy end of the duct, by the duct walls, is roughly equal to the intensity of the sound in the air at the duct entry, less the noise reduction figure for the duct walls. The importance of flanking transmission depends upon the situation in which the attenuating duct is used. In the case of a duct which extends outdoors, elimination of flanking transmission will not greatly reduce the total sound energy radiated by the duct walls unless the duct is very

long, for elimination of flanking cannot stop direct transmission through the walls at the high-intensity end. In general, however, it is good conservative design practice to include vibration breaks in the walls even of ducts which open outdoors. The longitudinal separation between successive breaks should be of the order of the duct width, where the duct has a roughly square or circular section.

The situation is somewhat altered when both the sound source and the point of reception are fully enclosed. For example, suppose that an air duct extends between two rooms which are in separate free-standing structures and which have acoustically opaque walls. The sound intensity level just outside the duct walls (outdoors), on the high-energy side, is approximately equal to the sound intensity level in the source room, less the noise reduction figure of the wall. If this sound intensity is somehow transmitted in the outside air to the exterior of the duct wall at the low-energy end, the passage of sound through the duct wall at this point will produce a level within the duct which is approximately equal to the intensity level in the source enclosure, less twice the duct-wall noise reduction figure. Therefore, when the source and the reception point are fully enclosed in separate structures, it is only necessary to require that the duct-wall noise reduction figure be somewhat greater than one-half of the total attenuation required of the duct. This relaxed requirement applies only when an adequate vibration break is provided in the duct walls; if flanking transmission is allowed, the duct-wall noise reduction must exceed the total attenuation requirement. It is not ordinarily necessary to provide more than one vibration break in the duct which connects separate enclosures, when the object is to reduce the noise level in the second enclosure only.

Telegraphing by Longitudinal Interior Members. The phenomenon of "telegraphing" is a form of flanking transmission which is attributable to passage of vibration along structural members within a duct. Longitudinal pipes, structural supports, and metallic facings for baffles and linings can act as paths for telegraphing. Structural members which might telegraph sound should be equipped with vibration breaks at suitable intervals. Usually it is sufficient to place the breaks at distances corresponding to 10 db increments of duct attenuation. Breaks may be omitted from any element which has the property of attenuating vibration rapidly; for example, a very thin metal facing in firm contact with an acoustical blanket may transmit little vibration.

Lateral Partitioning Behind Duct Linings. The space behind a duct lining constitutes a bypass for sound unless rigid, impervious baffles (partitions) are installed in such a way as to break the air space into short lengths. The length of duct between successive baffles should not exceed 1/10 wavelength at the frequency of maximum absorption if the calculated duct attenuation is to be realized. Usually the acoustical material in the duct lining is held in place by a perforated metal facing. The partitions can serve the auxiliary purpose of giving rigid support to the facing, in order to prevent destructive vibration. In vertical duct sections, the partitions also prevent the acoustical material from falling into one end of the lining space.

Extension of Parallel Baffle Treatment to the Walls. The predicted attenuation for parallel baffles can be obtained only when the treatment is continued in a consistent fashion across the entire interior of the containing duct. For example, the attenuation is greatly reduced when a set of baffles, on 8 in. center spacing, is installed in such a way as to leave an 8 in. air space adjacent to each wall of the containing structure.

The parallel baffle treatment can be extended to the walls in two different ways, as described below.

1. Install a baffle adjacent to each of the two walls at the sides of the array, even though this means that the outermost spaces between baffles will be less than the design value. Theoretically,the outermost baffles may have only one-half the thickness of acoustical material used in the other baffles in the set. In practice, it is often more convenient to allow the outer baffles to have full thickness, in order to obtain the advantages of uniform construction.

2. Install the set of baffles in such a way that, on each side, the center of the outermost baffle is separated from the side wall by one-half of the on-center spacing of adjacent baffles. The separation of the outermost baffles from the walls may be decreased below this value, but a larger separation is unacceptable.

Materials for Operation at Elevated Temperatures. For operation at elevated temperatures, as for example in exhaust ducts, mineral acoustical materials must generally be used. Mineral wools without organic binders (rock wool, Fiberglas) have been used at temperatures as high as 500°F, and some materials in this class have been used successfully at 1000°F.

Protection Against Wind Erosion. Special precautions are required when acoustical treatments are installed in high-speed air ducts. Tiles of mineral absorbing material suffer erosion and disintegrate gradually when directly exposed to high-speed air streams. A reasonably wind-resistant installation is obtained by placing loose mineral wool, or blankets not consolidated with binding material, behind a layer of glass cloth which is securely tied to the perforated metal facing. Further protection is obtained by placing a layer of 16-mesh fly screen between the facing and the cloth. When these precautions are observed, operation at maximum air velocities between 100 and 150 ft/sec is feasible. Porous-concrete block structures (See 12.3), when the blocks are adequately secured, will withstand similar velocities.

It is advisable to avoid air velocities exceeding 100 ft/sec. The problem of erosion protection becomes much more severe at higher velocities. Moreover, at higher velocities the noise power produced by the air flow (because of turbulence in the duct) may actually exceed the power passing through the attenuator from the original source. This self-noise problem has not been studied in detail, but there are measurements on record, for a porous-concrete installation, which suggest that the noise reduction is limited by the self-noise of wind flow. The air speed in this instance was 150 ft/sec.

Preventing Packing of the Acoustical Material. The unconsolidated mineral acoustical materials, which are suitable as a basis for high-velocity installations, are likely to pack in the lower part of the containing space as a result of vibration experienced in shipping or in service. The packing effect can be made negligible by dividing the containing space for the acoustical material into sections (each section about 2 ft in height). The sectioning may be accomplished by installing rigid partitions, and by forming the glass cloth protective covering into bags.

Avoidance of Chemical Action and Fouling. Chemical deterioration of acoustical linings and baffles is usually best avoided by use of mineral-wool acoustical materials, with facings made of aluminum or of stainless steel if necessary. While fouling of the acoustical material with soot and oil may occur in operation, permanent fouling difficulty can be avoided by periodic steam cleaning of mineral acoustical materials.

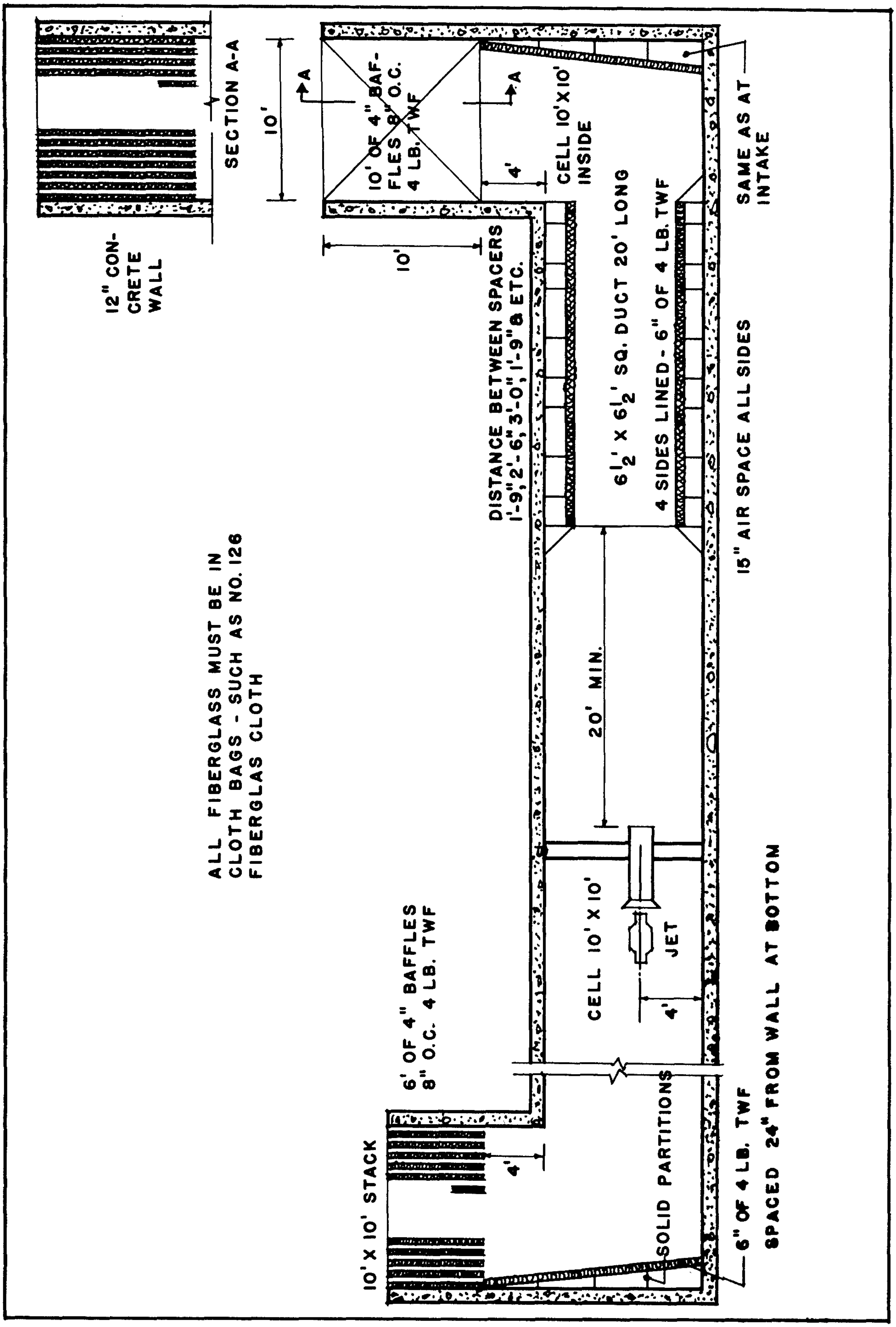
Avoidance of Vibration. There is considerable evidence that the most serious damage to linings and baffles which are subjected to high-speed or explosive air flow is attributable to structural vibration rather than directly to wind drag. The greatest damage results from vibration of the perforated metal facings. Usually adequate vibration reduction is obtained by fastening the facings securely, at frequent intervals, to the partitions within the acoustical material space. For air speeds of 100 ft/sec or greater, rigid lateral bracing between adjacent parallel baffles is required.

Unprotected edges of sheets of facing must be avoided. Edges should be sealed against wind entry. In the case of a baffle, adequate sealing is afforded by a metal strip crimped around the edge of the baffle. When wind is allowed to enter under the edges of a sheet of facing, strong vibration often results, with deterioration of the acoustical material and eventual tearing away of the facing.

Requirements for Perforated Facing. The perforated facing for linings or baffles must be essentially "acoustically transparent". An open area of 20 to 30 percent is normally required. The separation of adjacent perforations should not exceed approximately 1/2 in. The thickness of the material is governed by the conditions of operation. For wind speeds of about 100 ft/sec, the metal facing must be 20 gauge material or thicker. This minimum value is based upon an assumed 2-ft spacing for the supporting partitions.

Construction of Vibration Breaks. In general, a vibration break is a layer or section of material having a much lower stiffness per unit length than the structure into which the break is inserted. A layer of mastic, felt, asbestos, or rubber, inserted between sections of a heavy structure of wood, brick, concrete, or steel, is a vibration break. A flexible bellows or a length of bent rubber hose, connected between sections of piping, is also a satisfactory vibration break.

When it is structurally impracticable to insert a vibration break in a metal pipe, an acceptable substitute can sometimes be formed by packing the pipe in loose sand for a distance of several feet. Further references to vibration breaks are found in Chapters 11 and 13.


SECTION A-A
12" CON-
CRETE
WALL
10'
A
10' OF 4" BAF-
FLES 8" O.C.
4 LB. TWF
4'
CELL 10'X10'
INSIDE
SAME AS AT
INTAKE
DISTANCE BETWEEN SPACERS
1'-9", 2'-6", 3'-0", 1'-9" & ETC.
6½' X 6½' SQ. DUCT 20' LONG
4 SIDES LINED - 6" OF 4 LB. TWF
15" AIR SPACE ALL SIDES
ALL FIBERGLASS MUST BE IN
CLOTH BAGS - SUCH AS NO. 126
FIBERGLAS CLOTH
20' MIN.
CELL 10'X10'
JET
4'
6' OF 4" BAFFLES
8" O.C. 4 LB. TWF
10'X10' STACK
SOLID PARTITIONS
6" OF 4 LB. TWF
SPACED 24" FROM WALL AT BOTTOM

12.14 Numerical Example of a System for Control of Airborne Sound

This numerical example is concerned with the design of noise-control components for a hypothetical jet-engine test cell. The data do not represent recommendations for any actual problem. The cell, which has one intake stack and one exhaust stack, is illustrated in Fig. 12.22. It will be assumed that negligible sound transmission occurs through the walls of the cell. This assumption is not entirely unrealistic, because the thickness of test cell walls may be determined by non-acoustic considerations. The walls in the resulting design may have a greater transmission loss than the acoustic considerations require. It is further assumed that the total acoustic power enters the attenuating structure.

Suppose that no actual noise measurements on the sound source (the jet engine) are available. Estimated operating parameters of the engine are available, however. The overall power level and the spectrum may be obtained from the general design charts of Figs. 5.3 and 5.4. The information which is used to obtain acoustic data from the charts is shown below:

Fuel consumption, 5000 lb/hr
Static thrust, 4500 lb
S = (Thrust/fuel consumption) = 0.90 hr
$S^3 = 0.73$
Exhaust gas temperature at tail cone, 1650°R (Rankine)
Exhaust nozzle area, 286 sq in.
Ambient temperature, 60°F (520°R)
Exhaust nozzle (tail cone) diameter, 19 in.
Pressure of exhaust gas, 30 in. Hg

$$T - T_o = 1130^{\circ}F$$

$$W = (5.70)\, S^3 \,\frac{(T - T_o)^3}{T}\left[\frac{A}{286}\right] P = 3.68 \times 10^6 \text{ watts}$$

$$\frac{W}{W_o} \times \frac{T_o d_o}{Td} = \frac{3.68 \times 10^6 \times 1450 \times 19}{2.03 \times 10^6 \times 1650 \times 19} = 1.59$$

Figure 12.22

Hypothetical jet engine test cell and acoustical treatment. See numerical example of Sec. 12.14.

When the quantity $X = W\,[1 + (\frac{WT_o d_o}{W_o T d})^3] = 1.85 \quad 10^7$ watts

is used to enter the chart of Fig. 5.3, it is found that the predicted overall power level is

$$PWL = (168 \pm 3) \text{ db.}$$

For conservative practice, the maximum spectrum values of Fig. 5.4 are used. The resulting spectral distribution of power level is shown in Table 12.6.

It is also desirable to compute the air speeds in the intake and exhaust stacks. The air-fuel ratio is given as 50 to 1. The aspiration air is estimated as 3/2 of the combustion air. From these data the calculations below are made:

Combustion air consumption = 70 lb/sec
Aspiration air consumption = 105 lb/sec
Total mass flow through exhaust duct = 175 lb/sec.

The temperature of the mixed gas (exhaust and aspirated air) is now computed

$$T = \frac{1650 + 1.5 \times 520}{2.5} = 970^\circ R = 510^\circ F.$$

The specific volume of air at 30 in. pressure, 60°F is 13.2 ft^3/lb. Therefore, the volume flow in the exhaust stack is

$$U_{ex} = 175 \times 13.2 \times \frac{970}{520} = 4300 \text{ ft}^3/\text{sec}.$$

The volume flow in the intake stack is

$$U_{in} = 175 \times 13.2 = 2300 \text{ ft}^3/\text{sec}$$

The original area of each stack is 100 sq ft, but the open area will be reduced to about half when the acoustic treatment is installed. The approximate air speeds are

$$V_{ex} = 4300/50 = 86 \text{ ft/sec}$$

$$V_{in} = 2300/50 = 46 \text{ ft/sec}$$

These values are sufficiently small that they do not constitute an undue problem in the design of acoustical components.

The noise control requirements are now calculated by a modified form of the procedure illustrated in Sec. 10.4. The SPL at a distance of one mile is of interest. The maximum allowable values of SPL at this distance are shown in Table 12.6. The SPL spectrum which would be obtained at one mile, if the sound power were uniformly distributed over a hemisphere, is also indicated in the table. The SPL at one mile is obtained by subtracting from the power level the quantity $10 \log [2\pi(5280)^2]$, which is 82 db. The difference between SPL at one mile, as predicted in this basis, and the maximum allowable level, gives the noise reduction required for each octave band. It should be observed that an allowance for ground reflection has been included in the present calculations at the outset, by assuming hemispherical spreading of the signal.

The noise control requirements shown in Table 12.6 for the intake and for the exhaust are different. It is assumed that equal powers will be transmitted by the two paths to the point of observation. Because of the directionality of the sound source (see Chapter 5), the power level of the radiation into the exhaust duct is practically equal to the total power level. The noise-reduction required for the exhaust path is, therefore, 3 db in excess of the required overall noise reduction. (The 3 db correction allows the exhaust path to contribute one-half of the total transmitted power.) The level of the power transmitted to the intake end, according to Chapter 5, is less than the total power level in a given frequency band by (13 ± 2) db. For conservative practice, a value of 11 db is used for this correction. The noise required at the intake is therefore 11 db less than the exhaust requirements.

Table 12.7 shows the noise reduction for the various control measures proposed for the exhaust end. The directivity values used are the lower of the two possibilities of Sec. 12.10, because the flow of hot gas from the exhaust stack produces refraction which reduces the directivity. The proposed control scheme, which makes use of a combination of lined bend, lined duct, parallel baffles, and directional radiation, is adequate except at the highest frequency band. The noise reduction due to

TABLE 12.6

NOISE CONTROL REQUIREMENTS

Frequency Band	Source power level	SPL at one mile	Max. allowable SPL	Overall noise reduction required	Noise reduction required for exhaust	Noise reduction required for intake
cps	db	db	db	db	db	db
20-75	156	74	59	15	18	7
75-150	160	78	44	34	37	26
150-300	164	82	36	46	49	38
300-600	164	82	31	51	54	43
600-1200	162	80	27	53	56	45
1200-2400	160	78	23	55	60	49
2400-4800	155	73	20	53	58	47
4800-10000	154	72	18	54	57	46

TABLE 12.7

CALCULATED NOISE REDUCTION FOR EXHAUST

Frequency Band	Lined Bend attenuation	Attenuation of lined duct	Attenuation of baffles	Directivity index at 90°	Combined noise reduction	Deficiency
cps	db	db	db	db	db	db
20-75	5	9	2	3	19	-
75-150	13	15	4	6	38	-
150-300	18	15	9	7	49	-
300-600	20	15	24	9	68	-
600-1200	20	6	37	10	73	-
1200-2400	20	-	37	10	67	-
2400-4800	20	-	30	10	60	-
4800-10000	20	-	19	10	49	8

atmospheric absorption (Sec. 10.8), not shown in the calculations, amounts to some 15 db in the highest frequency band for a distance of one mile. When this is considered, the total noise reduction is adequate even in the highest band.

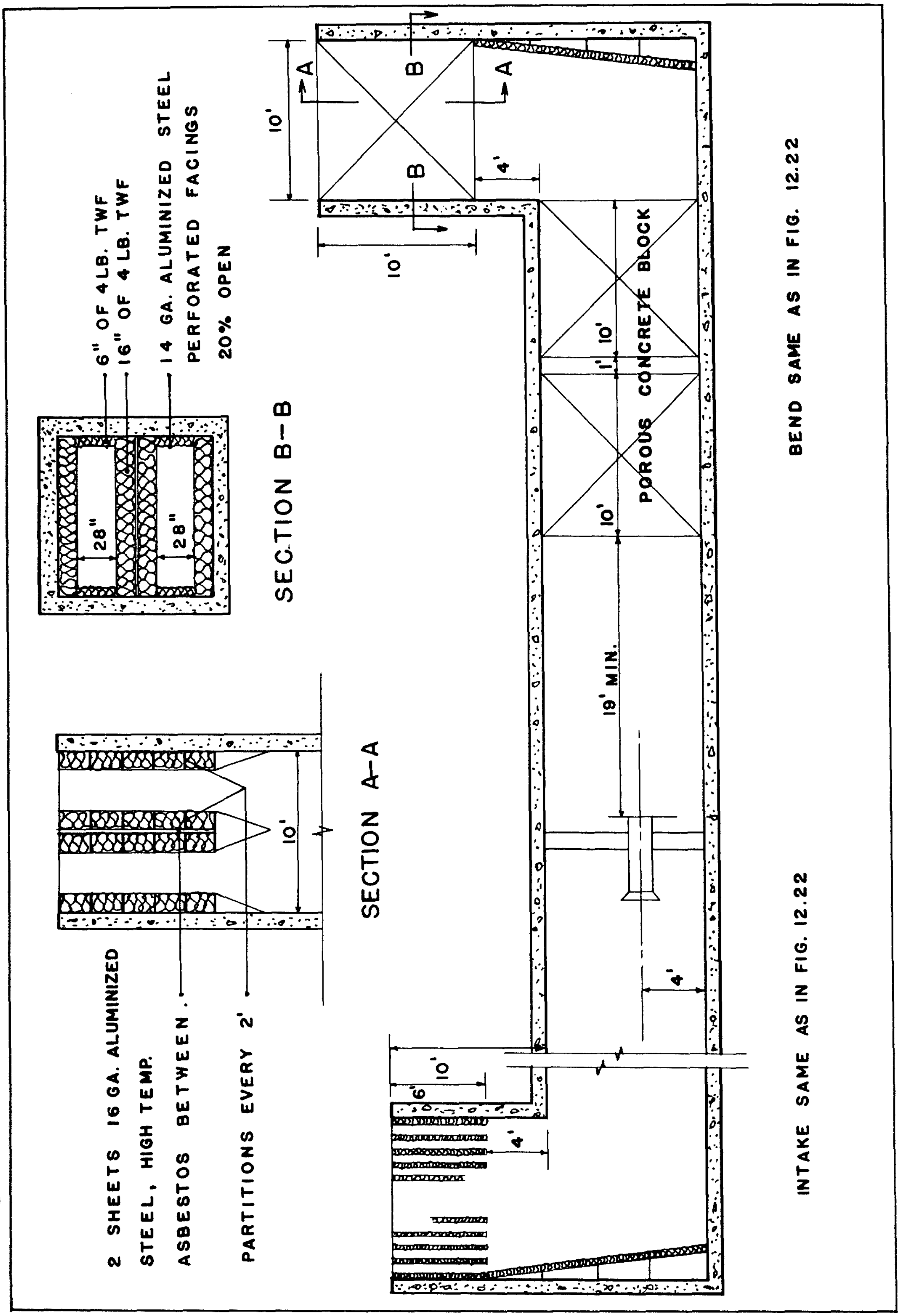
Table 12.8 shows the noise reductions for the control measures proposed for the intake end. Because of the less stringent requirements, no lined duct section is used in the intake. The higher values for directivity are used. The requirements are adequately met in all frequency bands.

TABLE 12.8

CALCULATED NOISE REDUCTION FOR INTAKE

Frequency Band	Lined bend attenuation	Attenuation of baffles	Directivity index at 90°	Combined noise reduction	Deficiency
cps	db	db	db	db	db
20-75	5	2	8	15	-
75-150	13	4	11	28	-
150-300	18	9	12	39	-
300-600	20	26	14	60	-
600-1200	20	40	15	75	-
1200-2400	20	40	15	75	-
2400-4800	20	32	15	67	-
4800-10000	20	20	15	55	-

It will be observed that the attenuation listed for the parallel baffles in the exhaust is slightly less than that listed for the baffles in the intake, although the length of the baffle treatment is 10 ft in the exhaust and only 6 ft in the intake. The explanation for the smaller attenuation per unit length assigned for the exhaust baffles is that a temperature correction has been introduced as described in Sec. 12.12. An allowance for the temperature effect has been made also in the lined duct data shown in the tabulation of exhaust noise reduction. The corrections have been made for a


2 SHEETS 16 GA. ALUMINIZED
STEEL, HIGH TEMP.
ASBESTOS BETWEEN.
PARTITIONS EVERY 2'
10'
SECTION A-A
6" OF 4LB. TWF
16" OF 4 LB. TWF
14 GA. ALUMINIZED STEEL
PERFORATED FACINGS
20% OPEN
28"
28"
SECTION B-B
A
A
B
B
10'
10'
4'
POROUS CONCRETE BLOCK
10'
1'
10'
19' MIN.
4'
10'
6'
4'
BEND SAME AS IN FIG. 12.22
INTAKE SAME AS IN FIG. 12.22

temperature of 500°F, which is the temperature of the exhaust gas entering the treatment. If the gas cools significantly in passing through the structure, the attenuation will exceed the calculated values.

An alternative exhaust treatment for the same test cell is shown in Fig. 12.23. The calculated performance data are shown in Table 12.9. An allowance has been made for the elevated temperature. For this reason the listed attenuation values for the lined duct section and for the porous concrete block section are less than the attenuations which would be given by these components at ordinary temperatures.

TABLE 12.9

CALCULATED NOISE REDUCTION FOR ALTERNATIVE EXHAUST TREATMENT

Frequency Band	Lined bend attenuation	Attenuation of concrete block	Attenuation of lined duct	Directivity index at 90°	Combined noise reduction	Deficiency
cps	db	db	db	db	db	db
20-75	5	6	10	3	24	-
75-150	13	9	12	6	40	-
150-300	18	12	20	7	57	-
300-600	20	14	25	9	68	-
600-1200	20	20	15	10	65	-
1200-2400	20	28	6	10	64	-
2400-4800	20	26	2	10	58	-
4800-10000	20	25	-	10	55	2

12.15 Attenuation of Sound Within Piping

As sound travels within piping, it will be attenuated by viscous friction at the inner sidewalls. The presence of branch

Figure 12.23
Hypothetical jet engine test cell of Fig. 12.22, with alternative acoustical treatment.

pipes will also tend to reduce the level in the main pipe. The attenuation in decibels per foot for sound passing down cylindrical piping is nearly negligible for frequencies much below 500 cps. Above 500 cps the attenuation may be estimated from the formula:

$$db/ft = 0.2 \ P/A$$

where P = perimeter of the pipe in inches
A = cross-sectional area of the pipe in sq in.

This formula has been found applicable to pipes with diameters between 48 and 120 inches. From field measurements it has been found that a reduction in sound level of 3 db results as a sound wave passes a side junction having a cross-sectional area equal to that of the principal pipe. It is assumed that the branch pipe has a length greater than a few wavelengths of sound. For shorter pipes the loss is usually less.

For smaller diameter circular pipes, one can use the relation given by Beranek 22/ :

$$db/ft = .0331 \quad f^{1/2}/R,$$

where f is the frequency in cycles/sec and R is the radius of the pipe in inches.

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CHAPTER 13

ROOMS AND SPECIAL ENCLOSURES

13.1 Introduction

A wide variety of noise-control problems can be solved by combinations of the techniques given in Chapters 11 and 12 for the control of structure-borne noise and for the control of airborne noise. Perhaps the most frequent problem which calls for a combination of these techniques is that of providing a suitable acoustical environment in a room. The present chapter is intended to show how the various noise control principles are integrated to solve this particular problem. The basic engineering information for noise control which has already been presented will not be repeated. A general concept of the treatment of the room as a unit will be developed, and several specific procedures and treatments which are particularly useful in the room problem will be presented.

The present discussion deals largely with the control of noise levels existing within rooms as a result of external sound sources. The question of calculating the proper amount and distribution of absorbing material for control of sound originating within the room is not discussed in detail, although the methods of this chapter are directly applicable. Adequate published information is available to guide the designer in these problems. 1/ (See also the references to the Bulletin of the Acoustical Materials Association, Knudsen and Harris, and Less Noise, Better Hearing in Sec. 2.5 of this manual.)

13.2 Engineering Description of Sound Sources Affecting a Room

A recommended standard procedure for quantitative description of noise sources affecting a room is to derive an equivalent value of source power, or source power level, for each noise source. By an equivalent source power level is meant a quantity which can be inserted into Eqs. (3.10) or (3.11) to find the sound pressure level (SPL) within a room, or in other words, the power level which would have to be developed by a source located within the room to produce the actual SPL. The value of power which corresponds to the equivalent PWL is the equivalent source power.

As an example of a typical procedure when sound is transmitted through a wall, consider the following case:

A room wall, having dimensions 10 ft by 20 ft, gives a 30 db noise reduction in the 300-600 cps band. The external SPL in this band is 110 db. The equivalent source power within the room is to be found.

The SPL just inside the wall is given approximately by

$$SPL_{internal} = SPL_{external} - \text{Noise Reduction},$$

so that the SPL just inside the wall is 80 db. The sound pressure just inside the wall is assumed to be connected with the power per unit area of wall by Eq.(2.1), which gives the intensity (power per unit area) in a plane wave. Suppose that ρc in cgs units is 40. Then, by Eq. (2.1), the power per unit area of wall at the inside surface is 0.1 erg/sec-cm^2, or 1.0×10^{-8} watt/cm^2. Finally, the total power in the 300-600 cps band, for the wall of area 200 sq ft (or 1.86×10^5 sq cm), is $10^5 \times 1.86 \times 10^{-8} = 1.86 \times 10^{-3}$ watt.

A typical procedure when sound enters the room through an opening is illustrated in the following example: The room is connected to an external sound field, of SPL 92 db in the 75-150 cps band, by an air-conditioning duct which has an attenuation of 18 db in this band. The duct opening has an area of 2 sq ft. The equivalent source power in the room is to be found.

The calculation follows exactly the procedures of the previous example. The SPL in the room, next to the duct opening, is 92-18 or 74 db, in the 75-150 cps band. The corresponding rms sound pressure is 1.0 dynes/cm^2. Therefore the power due to the entire opening, in the 75-150 cps band, is 1830 cm^2 $\times$ 2.5×10^{-9} watt/cm^2, or 4.6×10^{-6} watt.

The total values of equivalent source power or power level in each frequency band are converted to values of SPL within the room by the use of Eqs. (3.10) and (3.11), and Fig. 3.3. When there is nearly uniform distribution of power over most of the wall area of the room, the SPL within the room will be nearly uniform. Then a single value of the SPL computed from Eq. (3.11), which applies in the case of a duffuse sound field, describes the sound level in the room adequately.

When the source is concentrated in one part of the room, the SPL falls off with increasing distance from the source and approaches a limiting uniform value when the distance exceeds a critical value. A good description of the distribution of SPL

in this case is obtained by considering the source as a point, and applying Eq. (3.11) or Fig. 3.3. The value of SPL far from the source is the same as that which would be obtained with a distributed source of the same power; therefore, if a precise description of the behavior of the SPL near the source is not required, it is sufficient in practice to use Eq. (3.11) even for a concentrated source.

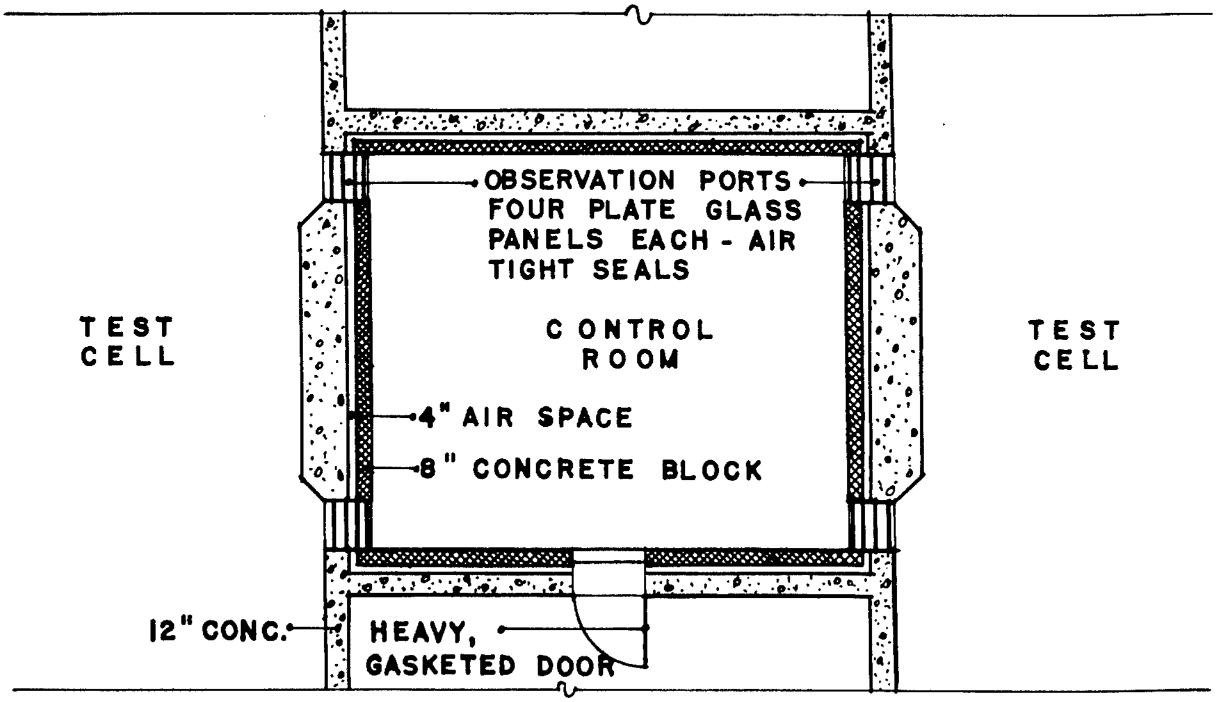
13.3 Planning for Room Noise Control

Usually noise control measures for a room which has been already constructed, without acoustical planning, are much more expensive than the measures which can be used when noise control is made a part of the original design. The most efficient solution to a noise control problem is obtained when both location and structure of rooms are examined, with reference to noise, in the first stages of design.

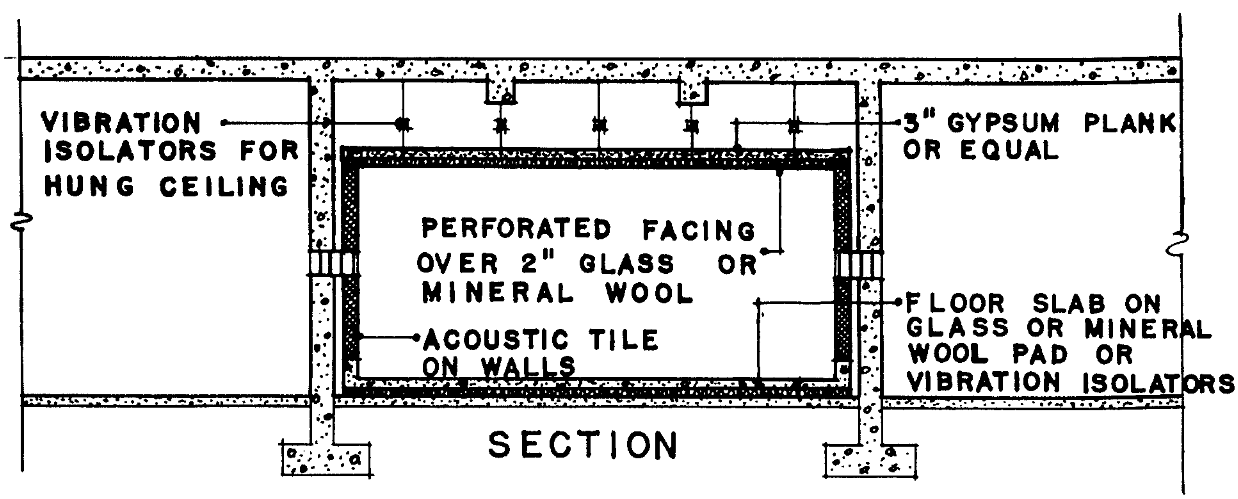
Rooms in which low noise levels are required should be located as far as possible from sources of noise. For example, it is preferable not to place an office in a building which contains a heavy machine shop; also, the windows of the office should not face an intense noise source. If an office must be placed in the same building as a heavy machine shop, then the office should if possible be separated from the shop by a dead area (for example, by a storeroom). Ventilating machinery and other noisy service facilities should be isolated from quiet areas. If reasonable separation of quiet areas from noise source can be achieved in the initial design of a building or group of buildings, special structural features for sound isolation may not be required, or may be no more complicated than a vibration break in the framework of a building.

In certain instances, it is necessary to control the noise level in a room which cannot be removed from the noise source. An example is the control room of an engine test cell. Planning for noise control should nevertheless be initiated in the early design stage, because special structural features will usually be required, and ordinarily these can be provided at less cost as original construction than as modifications.

The original planning for noise control should if possible be quantitative, and have its basis in measured values of source power levels. If intense noise sources are involved for which no power level data are available, the expenditure required to obtain measurements on similar sources will generally be justifiable. Often the most convenient procedure is to set up a



PLAN



SECTION

tentative structural design, estimated to be approximately suitable, and then, by acoustical analysis of this specific structure, to arrive at final design details for acoustical control. A numerical example of noise control calculations for a room is given in the next section.

13.4 Example of Sound Control Calculations for a Room

The specific example given below of sound control for a room concerns a control room located between two jet-engine test cells. This problem requires particularly rigorous sound-reduction measures. The maximum allowable values of SPL in the cell are shown in Table 13.1. These values represent an arbitrary choice for the present example. For simplicity, intermediate steps in the design will be omitted, and the example will consist of the calculations which verify the proper performance of the final design.

The control room structure is illustrated in Fig. 13.1. The following features have particular significance in the acoustical design:

1. The room is situated between two test cells. The axis of each cell is parallel to the axis of the room. The cells may be operated simultaneously.

2. Each cell has a total open area of 2200 sq ft. The openings are located at the ends of the cells and radiate horizontally.

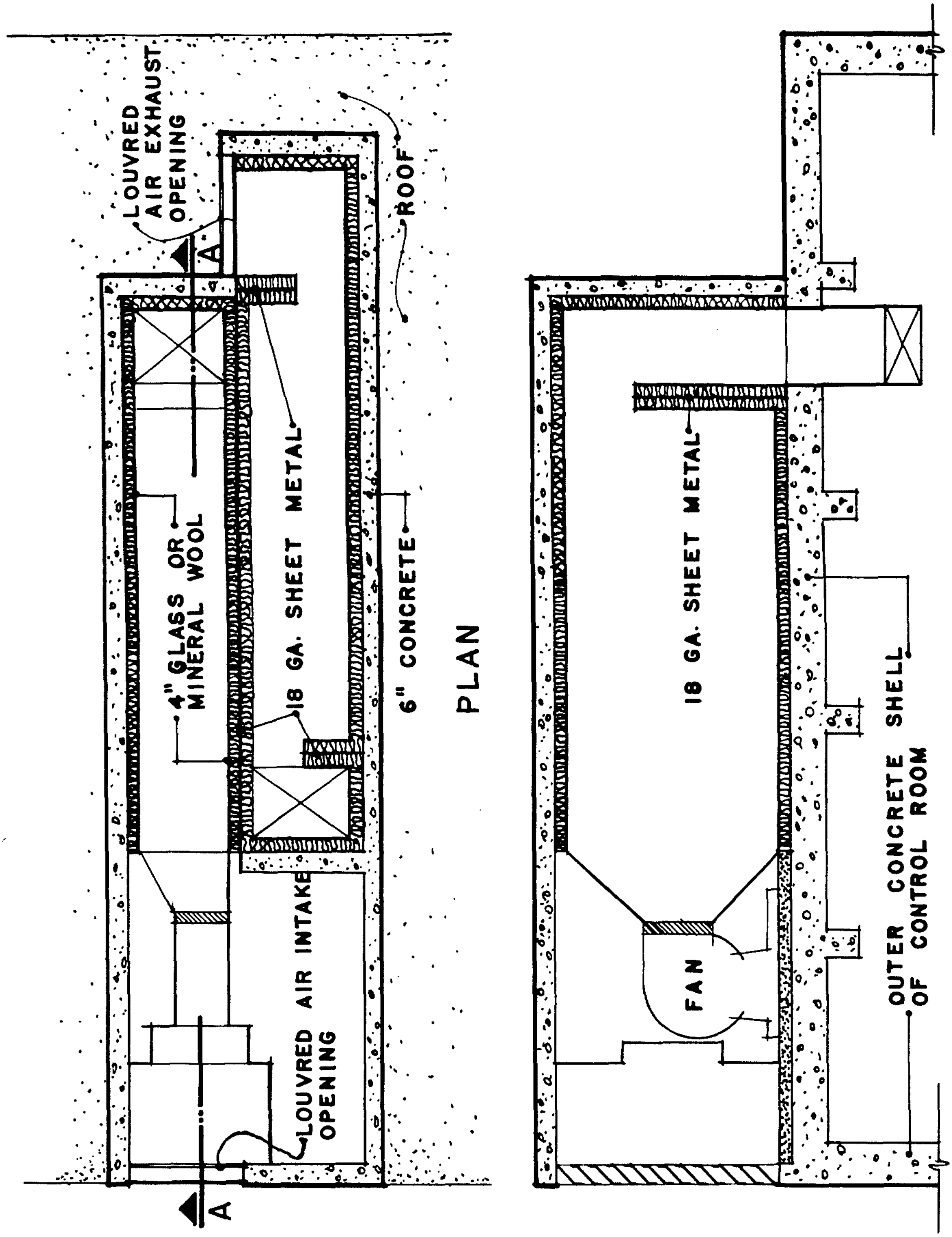
3. The control room has a double wall, the outer wall consisting of concrete 12 in. thick, and the inner wall of 8 in. solid concrete block. The air space is 4 in.

4. The roof and ceiling construction is as follows: The test cell and the control room are each roofed with 12 in. concrete slab. The test cell has no separate ceiling panel. The control room ceiling panel, suspended 2 ft below the roof slab, is 3 in. gypsum plank, on which the acoustic material for the ceiling is mounted.

5. The floor slab of the control room is isolated from the concrete of the surrounding structure by a blanket of glass wool or by suitable vibration isolators. Floor vibration may be therefore neglected.

Figure 13.1

Plan and elevation of the test cell used in the sample sound control calculations.


LOUVRED AIR EXHAUST OPENING
ROOF
A
4" GLASS OR MINERAL WOOL
18 GA. SHEET METAL
6" CONCRETE
LOUVRED AIR INTAKE OPENING
A
PLAN
18 GA. SHEET METAL
FAN
OUTER CONCRETE SHELL OF CONTROL ROOM
SECTION A-A

6. Each observation window is composed of four separate glass panels (see Fig. 13.7), two in the test cell wall and two in the control room wall. The intervening air spaces are approximately 6 in. each. The first glass panel (on the test cell side) is 3/4 in. thick; the others are 1/2 in. thick.

7. Neoprene seals are used at all points of contact between the steel window frames and the concrete structure.

8. The doors are equipped with hollow molded neoprene gaskets as illustrated in Fig. 13.8.

9. Heating and ventilating fans are mounted on rubber-in-shear supports with resonant frequency less than one-third the blade passage frequency. Vibration from the fans may be neglected.

10. The intake and exhaust ducts of the ventilating system, illustrated in Fig. 13.2, are acoustically treated. Attenuation data will be presented in the calculations.

11. The electrical connections to the test cell are carried through the walls in acoustically sealed conduits and lie in a treated control duct in the test cell. The arrangement is illustrated in Fig. 13.3. It is assumed that this treatment reduces noise transmission at the cable entries to negligible values.

12. The acoustical absorbing material within the room consists of the following:

Ceiling treatment, 2 in. glass or mineral wool blanket with perforated metal facing, 240 sq ft.

Wall treatment, 3/4 in. acoustic tile, 160 sq ft.

The total surface area is 1090 sq ft. The computed values of the room factor R are given in Table 13.1.

Design values are given for the octave-band SPL which will exist within a test cell, and just outside the openings of the test cell. These are listed in Table 13.2

Figure 13.2

Detail of the intake and exhaust ducts of the ventilating system for the control room.

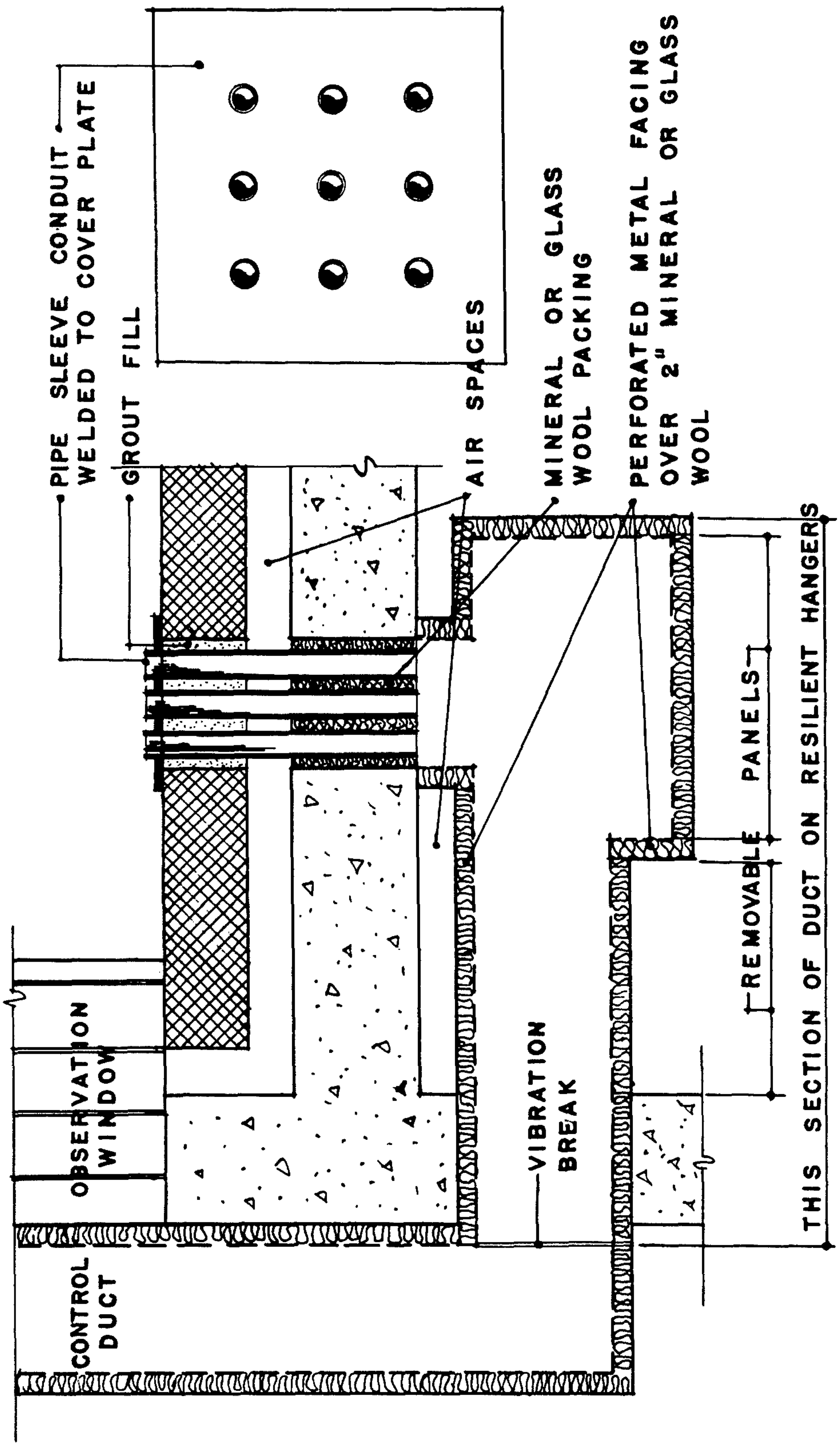

PIPE SLEEVE CONDUIT WELDED TO COVER PLATE
GROUT FILL
AIR SPACES
MINERAL OR GLASS WOOL PACKING
PERFORATED METAL FACING OVER 2" MINERAL OR GLASS WOOL
THIS SECTION OF DUCT ON RESILIENT HANGERS
REMOVABLE PANELS
VIBRATION BREAK
OBSERVATION WINDOW
CONTROL DUCT

TABLE 13.1

ROOM FACTOR R FOR CONTROL ROOM

Frequency Band cps	R, sq ft
20-75	53
75-150	102
150-300	259
300-600	735
600-1200	1050
1200-2400	895
2400-4800	778
4800-10000	715

TABLE 13.2

GIVEN VALUES OF SPL, ONE CELL OPERATING

Frequency Band cps	SPL in Cell	SPL outside openings	Allowable SPL at Operator's Location
20-75	131	114	93
75-150	135	99	83
150-300	138	88	75
300-600	140	82	68
600-1200	139	80	63
1200-2400	135	80	59
2400-4800	131	80	57
4800-10000	125	80	56

Figure 13.3
Detail of the control duct used to introduce electrical connections from the test cell to the control room.

The values for SPL outside the cell are based upon allowable levels specified for a residential neighborhood 2000 ft distant. The SPL are obtained from these allowable levels by a simple inverse-square correction for distance. The design of acoustical treatment for the intake and exhaust of the cell which will insure that the indicated levels are actually the ones existing outside the cell, is a separate problem. A problem of this nature is discussed in Sec. 12.14.

The plans are studied carefully for possible paths by which noise may be transmitted from the test cell to the control room. The paths which are selected as requiring examination are listed below. The observation windows have a sufficiently larger transmission loss so that it is not necessary to consider them apart from the entire separating wall.

1. Test cell noise passing through the double wall structure made up of the test cell wall and the control room wall with intervening air space.

2. Test cell noise passing through the test cell wall into the duct space and then through the gypsum plank ceiling into the room.

3. Test cell noise passing through the test cell wall, into the duct space, then through the wall of the ventilation ducts (both exhaust and intake), and finally through the duct opening into the room.

4. Outside noise passing through the test cell wall, into the duct space, then through the wall of the ventilation ducts (both exhaust and intake), and finally through the duct opening into the room.

5. Outside noise leaking through the concrete sidewalls of the baffle casing (see Fig. 13.3) and then through the duct openings into the room.

6. Outside noise traveling along the exhaust ventilating duct into the room.

7. Outside noise traveling along the intake duct into the room.

Noise transmitted by each path is now expressed in terms of equivalent source power in the control room, as illustrated in Sec. 13.2. Wherever sound transmission through a wall is

considered in this particular problem, the noise reduction for the wall is taken to be the standard TL plus 6 db. This is the effective noise reduction value when the receiving room (control room) is "dead" (see Sec. 11.2). From Table 13.1 it can be seen that the high absorption in the room renders it relatively "dead", particularly in the middle and high frequency bands. Where experimental information was not available, the TL values in the following tables were derived from the mass law as corrected for random incidence (see Fig. 11.2).

Path 1. Test cell noise passing into the control room through the intervening double wall structure (12 in. concrete wall, 4 in. air space, and 8 in. solid concrete block wall) is derived as shown in Table 13.3.

TABLE 13.3

TEST CELL NOISE TRANSMITTED TO CONTROL ROOM THROUGH ONE SIDEWALL

Frequency Band	Sound Pressure Level in Cell	Noise Reduction of Double Wall	SPL Inside Control Room near Wall	Acoustic Power at Face of Wall	Total Acoustic Power at Wall
cps	db	db	db	watts/cm^2	watts
20-75	131	68	63	2.0×10^{-10}	3.2×10^{-5}
75-150	135	70	65	3.2×10^{-10}	5.0×10^{-5}
150-300	138	72	66	4.0×10^{-10}	6.3×10^{-5}
300-600	140	75	65	3.2×10^{-10}	5.0×10^{-5}
600-1200	139	80	59	0.8×10^{-10}	1.3×10^{-5}
1200-2400	135	87	48	0.6×10^{-11}	0.95×10^{-6}
2400-4800	131	87	44	2.5×10^{-12}	4.0×10^{-7}
4800-10000	125	87	38	0.6×10^{-12}	0.95×10^{-7}

Path 2. Noise passing through the 12 in. concrete test cell wall and then through the duct space above the control room ceiling and then through the 3 in. gypsum plank ceiling into the control room. Values are then derived as shown in Table 13.4.

Path 3. The tabulation of Table 13.5 is for the test cell noise which passes through the cell wall into the duct space, and through the walls of the ventilation ducts, both exhaust and

intake, and finally through the duct openings into the room. The ducts are made of 18 gauge steel coated on the inside with 1/2 in. thick Fiberglas Coated Duct Insulation or its equivalent and covered on the outside with four layers of 1 in. Fiberglas PF 316, or its equivalent. (1 lb per cu ft)

TABLE 13.4

TEST CELL NOISE TRANSMITTED TO THE CONTROL ROOM THROUGH THE CEILING (AREA = 240 SQ FT)

Frequency Band	SPL in Cell	Noise Reduction for 12 in. Concrete plus Plenum plus 3 in. Gypsum	SPL Inside Room at Ceiling	Acoustic Power at Ceiling	Total Power at Ceiling
cps	db	db	db	watts/cm^2	watts
20-75	131	61	70	1.0×10^{-9}	2.2×10^{-4}
75-150	135	77	58	0.6×10^{-10}	1.4×10^{-5}
150-300	138	88	50	0.1×10^{-10}	0.2×10^{-5}
300-600	140	100	40	0.1×10^{-11}	0.2×10^{-6}
600-1200	139	100	39	0.8×10^{-12}	1.7×10^{-7}
1200-2400	135	100	35	0.3×10^{-12}	0.7×10^{-7}
2400 4800	131	100	31	0.1×10^{-12}	0.3×10^{-7}
4800-10000	125	100	25	0.3×10^{-13}	0.7×10^{-8}

Path 4. By this path, outside noise may enter the control room after passing through the air intake on the roof of the building, through the sheet metal partition which separates the intake and exhaust ducts of the ventilating system, and through the exhaust duct opening into the room. As a basis for the path (4) calculation it is necessary to estimate the noise levels on the roof of the control-classroom section of the building. This estimation is tabulated in Table 13.6.

TABLE 13.5

TEST CELL NOISE TRANSMITTED TO THE CONTROL ROOM THROUGH THE VENTILATION OPENINGS

(Area = 8 sq ft)

Frequency Band	SPL in Test Cell	Composite Noise Reduction of concrete plus sheet metal plus wrapping	SPL at Duct Exit to Control Room	Acoustic Power at Ceiling	Total Power at Ceiling
cps	db	db	db	watts/cm^2	watts
20-75	131	47	84	2.5×10^{-8}	1.9×10^{-4}
75-150	135	61	74	2.5×10^{-9}	1.9×10^{-5}
150-300	138	72	66	4.0×10^{-10}	3.0×10^{-6}
300-600	140	86	54	2.5×10^{-11}	1.9×10^{-7}
600-1200	139	96	43	2.0×10^{-12}	1.5×10^{-8}
1200-2400	135	100	31	0.1×10^{-12}	2.4×10^{-9}
2400-4800	131	100	31	0.1×10^{-12}	1.0×10^{-9}
4800-10000	125	100	25	0.3×10^{-13}	2.4×10^{-10}

In the following table, which shows the levels arriving by path (4), it is assumed that the cells on each side of the control room are operating so that the levels in the last column of Table 13.6 are applicable.

Path 5. This contribution to the noise levels in the control room arises from outside noise leaking through the 6 in. concrete sidewalls of the roof ventilation ducts and then directly into the exhaust openings into the room. The levels produced by this source are shown in Table 13.8

TABLE 13.6

ESTIMATED LEVELS ON THE ROOF OF THE CONTROL ROOM

Frequency Band	Estimated Levels outside the Cell Openings (Table 13.2)	Loss due to Directivity for a 14x14 ft Opening	Estimated Loss due to Distance	Estimated SPL on Roof- One Cell Operating	Estimated SPL on Roof Two Cells Operating
cps	db	db	db	db	db
20-75	114	1	6	107	110
75-150	99	4	6	89	92
150-300	88	11	6	71	74
300-600	82	17	6	59	62
600-1200	80	23	6	51	54
1200-2400	80	25	6	49	52
2400-4800	80	25	6	49	52
4800-10000	80	25	6	49	52

TABLE 13.7

OUTSIDE NOISE TRANSMITTED TO THE CONTROL ROOM THROUGH THE DIVIDING SHEET METAL PARTITION AND THEN THROUGH THE EXHAUST DUCT OPENING (AREA = 6 SQ FT)

Frequency Band	Estimated SPL on Roof	Composite Noise Reduction	SPL at Duct Opening in Ceiling	Acoustic Power at Ceiling	Total Power at Ceiling
cps	db	db	db	watts/cm^2	watts
20-75	107	13	94	2.4×10^{-7}	1.3×10^{-3}
75-150	89	16	73	2.0×10^{-9}	1.1×10^{-5}
150-300	71	35	36	0.4×10^{-12}	2.2×10^{-9}
300-600	59	41	18	0.6×10^{-14}	3.4×10^{-11}
600-1200	51	46	5	-	-
1200-2400	49	51	1	-	-
2400-4800	49	56	0	-	-
4800-10000	49	56	0	-	-

TABLE 13.8

OUTSIDE NOISE TRANSMITTED THROUGH THE VENTILATION CASING INTO THE CONTROL ROOM

Frequency Band	Estimate SPL on Roof	Noise Reduction of 6 in. Concrete Wall	SPL at Duct Opening in Ceiling	Acoustic Power at Ceiling	Total Power at Ceiling (Area = 6 sq ft)
cps	db	db	db	watts/cm^2	watts
20-75	107	36	71	1.3×10^{-9}	0.7×10^{-5}
75-150	89	45	44	2.6×10^{-12}	1.4×10^{-8}
150-300	71	50	21	1.3×10^{-14}	1.0×10^{-10}
300-600	59	56	3	-	-
600-1200	51	61	0	-	-
1200-2400	49	66	0	-	-
2400-4800	49	70	0	-	-
4800-10000	49	70	0	-	-

Path 6. Outside noise may be transmitted to the control room directly through the exhaust end of the ventilating duct. The levels produced by this source are shown in Table 13.9.

TABLE 13.9

OUTSIDE NOISE TRANSMITTED DIRECTLY THROUGH THE EXHAUST DUCT INTO THE CONTROL ROOM

Frequency Band	Estimated SPL on Roof	Duct Attenuation	SPL at Duct Opening in Ceiling	Acoustic Power at the Ceiling	Total Power at the Ceiling (Area = 6 sq ft)
cps	db	db	db	watts/cm^2	watts
20-75	107	16	91	1.2×10^{-7}	0.7×10^{-3}
75-150	89	21	68	0.6×10^{-9}	3.4×10^{-6}
150-300	71	36	35	0.4×10^{-12}	2.2×10^{-9}
300-600	59	50	9	1.0×10^{-16}	0.6×10^{-12}
600-1200	51	50	1	-	-
1200-2400	49	50	0	-	-
2400-4800	49	50	0	-	-
4800-10000	49	50	0	-	-

Path 7. Outside noise may be transmitted to the control room directly through the intake end of the ventilating duct. The levels produced by this source are derived in Table 13.10.

TABLE 13.10

OUTSIDE NOISE TRANSMITTED DIRECTLY THROUGH THE INTAKE DUCT INTO THE CONTROL ROOM

Frequency Band	Estimated SPL on Roof	Duct Attenuation	SPL at Duct Opening in Ceiling	Acoustic Power at Ceiling	Total Power at the Ceiling (Area = 2 sq ft)
cps	db	db	db	watts/cm^2	watts
20-75	107	14	93	2.0×10^{-7}	3.8×10^{-4}
75-150	89	18	71	1.2×10^{-9}	2.2×10^{-6}
150-300	71	31	40	1.2×10^{-12}	2.2×10^{-9}
300-600	59	50	9	1.0×10^{-16}	1.8×10^{-13}
600-1200	51	50	1	-	-
1200-2400	49	50	0	-	-
2400-4800	49	50	0	-	-
4800-10000	49	50	0	-	-

Finally a condensed tabulation is made of the acoustic powers for all paths and the total power for each octave band is computed. This is shown in Table 13.11. The total power is then used in Eq. (3.11) to arrive at the values of SPL in the control room. When c is 37 in cgs units, Eq. (3.11) becomes

$$SPL = 76.5 + 10 \log P - \log R \qquad (13.1)$$

where P is the power in microwatts of the sources operating in the room and R is the room factor in sq ft. If PWL denotes the power level of the sources re 0.1×10^{-13} watt, the relation becomes

$$SPL = 6 + PWL - 10 \log R \qquad (13.2)$$

TABLE 13.11

ACOUSTIC POWER IN MICROWATTS

TRANSMITTED TO CONTROL ROOM THROUGH DIFFERENT PATHS

Frequency Band	Path 1	Path 2	Path 3	Path 4	Path 5	Path 6	Path 7	Total Power in Control Room
20-75	32	220	190	1300	7	700	380	2830
75-150	50	14	10	11	0.01	3.4	2.2	100
150-300	63	2	3	-	-	-	-	68
300-600	50	-	-	-	-	-	-	50
600-1200	13	-	-	-	-	-	-	13
1200-2400	0.9	-	-	-	-	-	-	0.9
2400-4800	0.4	-	-	-	-	-	-	0.4
4800-10000	0.09	-	-	-	-	-	-	0.09

The values of R are given in Table 13.1. The final computed values or the SPL are given in Table 13.12.

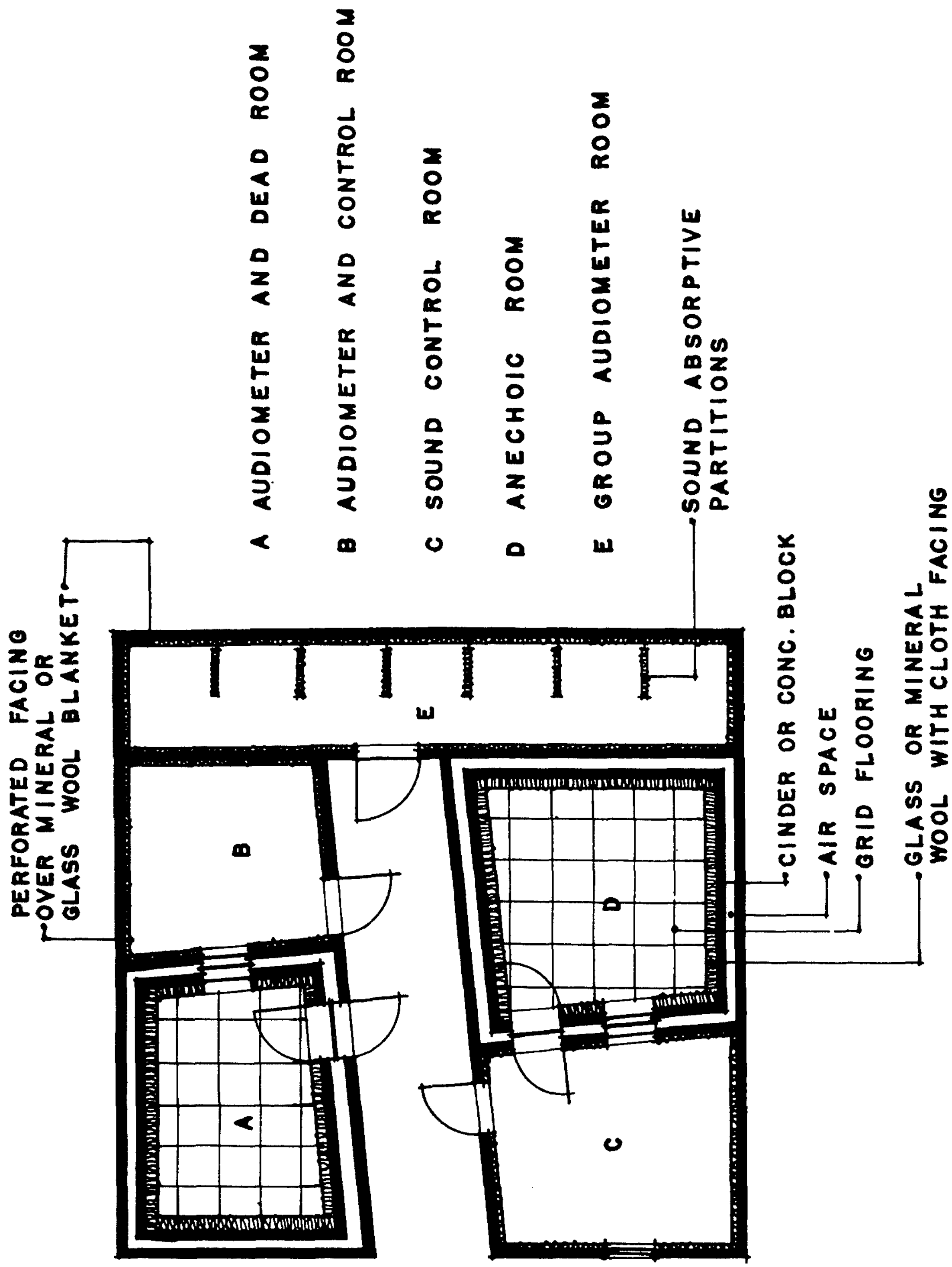
TABLE 13.12

COMPUTED SPL IN THE CONTROL ROOM

Frequency Band cps	SPL, One Cell Operating	SPL, Two Cells Operating
20-75	94	97
75-150	76	79
150-300	71	74
300-600	65	68
600-1200	57	60
1200-2400	47	50
2400-4800	44	47
4800-10000	38	41

The calculations give the values for one cell operating. The values for two cells are obtained by adding 3 db (corresponding to power doubling). Comparison of the spectrum for two-cell operation with the allowable levels of Table 13.2. shows that the requirements are met except for a deficiency of 4 db in the lowest frequency band.

The calculations were made on the assumption of a diffuse sound field within the control room, and therefore would be expected to apply in general only at distances greater than some 10 ft from concentrated sources. It happens in the present example however, that the concentrated sources (the duct openings) in no case contribute the larger part of the power. As a consequence, the SPL in the room is somewhat in excess of that which would exist at the duct openings in the absence of other sources, and an increase of SPL near the duct openings will not be found.


A AUDIOMETER AND DEAD ROOM
B AUDIOMETER AND CONTROL ROOM
C SOUND CONTROL ROOM
D ANECHOIC ROOM
E GROUP AUDIOMETER ROOM
SOUND ABSORPTIVE PARTITIONS
PERFORATED FACING OVER MINERAL OR GLASS WOOL BLANKET
CINDER OR CONC. BLOCK
AIR SPACE
GRID FLOORING
GLASS OR MINERAL WOOL WITH CLOTH FACING
A
B
C
D
E

13.5 Isolating Enclosure for a Sound Source Within a Room

The procedures illustrated in Sec. 13.4 may be applied directly in the design of a housing or enclosure to reduce room noise levels which arise from a source within a room. For example suppose that noise resulting from a motor-generator set in the room is to be reduced to a specified acceptable level. First a sound survey is made to establish the power level spectrum of the source. The principles of this survey are discussed in Chapter 3. It will usually be desirable to choose a large number of microphone positions which are near enough to the source to justify the assumption of free-field conditions, although the power level can be derived from space-average SPL in a reverberant room when the acoustic properties of the room are well known, and when there are no interfering noise sources.

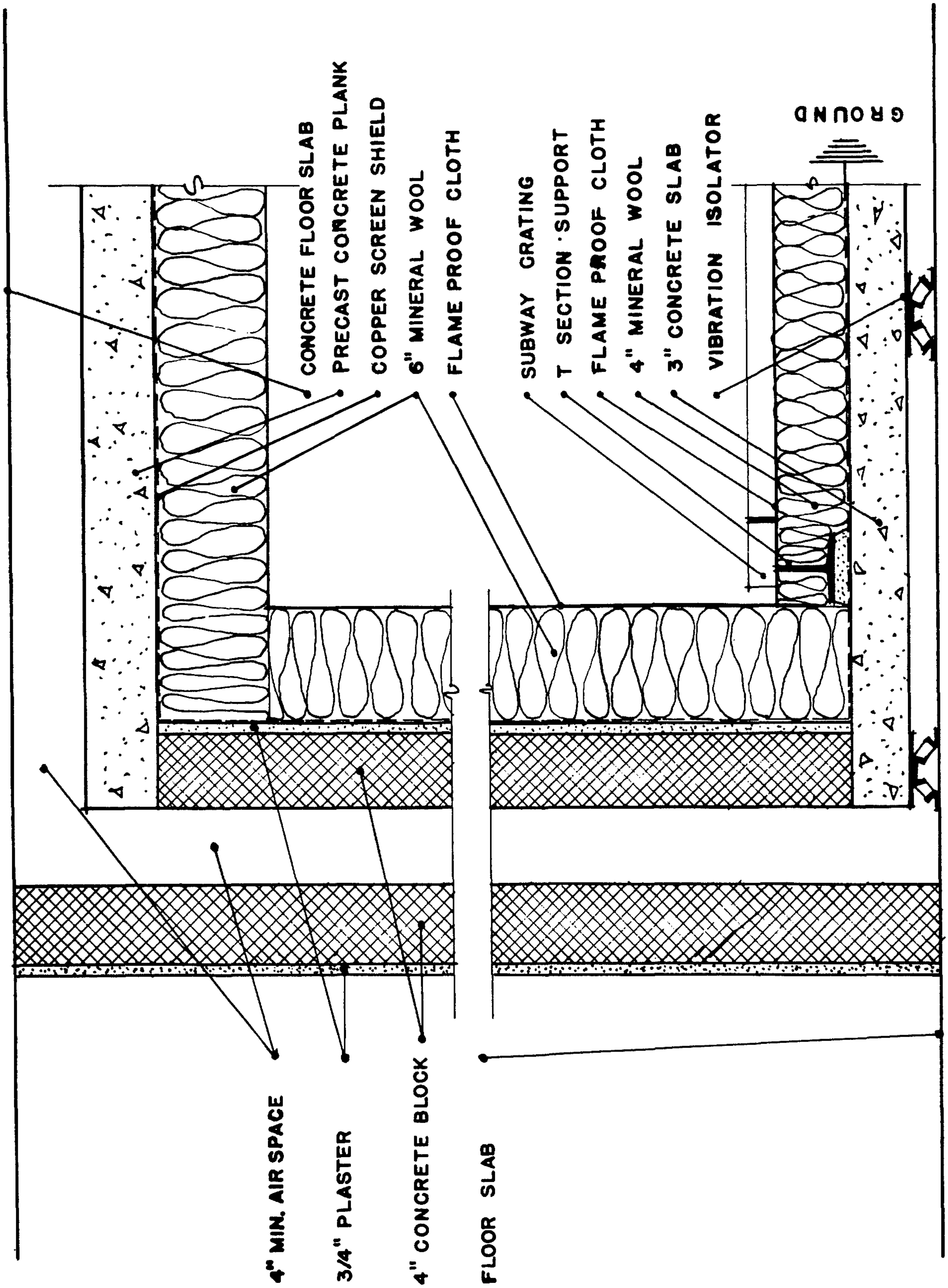
When the housing is in place, the inside space assumes the role of the test cell in Sec. 13.4, and the remainder of the room assumes the role of the control room in the previous example. The SPL spectrum within the enclosure can be computed from the known power level spectrum when a decision is reached as to the amount and kind of sound-absorbing material which will be placed within the enclosure. The required transmission loss for the housing can be computed if the acoustical properties of the room are known when acceptable values for SPL in the room are specified. Alternatively, a tentative design for the housing can be established and SPL in the room then computed exactly as in Sec. 13.4.

13.6 Design for Audiometry Unit

In the design of rooms to obtain the desired medical results from audiometric tests, often the greatest single criterion to be met is that the room must be integrated into an existing structure. In addition, non-reflective walls are required for free-field audiometry. With this in mind, a useful group unit is presented in this section. The unit utilizes acoustical principles, methods and materials developed as a result of experience in many types of buildings and it may be incorporated into the design of any future structure or may be built in an existing building by removal of existing partitions.

Figure 13.4

Audiometry unit, in which hearing tests are made.


CONCRETE FLOOR SLAB
PRECAST CONCRETE PLANK
COPPER SCREEN SHIELD
6" MINERAL WOOL
FLAME PROOF CLOTH
SUBWAY GRATING
T SECTION SUPPORT
FLAME PROOF CLOTH
4" MINERAL WOOL
3" CONCRETE SLAB
VIBRATION ISOLATOR
GROUND
4" MIN. AIR SPACE
3/4" PLASTER
4" CONCRETE BLOCK
FLOOR SLAB

This clinical acoustic unit is shown in Fig. 13.4. The Anechoic Room ("dead room") and the Sound Control Room are designed to be used together. In the control room, electro-acoustic equipment is operated by a technician; the patient sits in the "dead room" while the tests are made. The Audiometer and Control Room and the Audiometer and "dead room" have dual functions: they can be used together for the same purpose as the pair described above, but they are intended primarily to be used singly for monaural audiometer tests. This explains the different arrangement of the doors. The Group Audiometry room is used independently and accommodates up to six patients. The corridor is so placed as to act as a soundblock for any of the rooms.

In the planning of this group unit, factors mentioned previously, such as segregation from noisy areas, are assumed to have been recognized. Examination of Fig. 13.4 illustrates the location of doors and slant walls to assist in elimination of flutter echoes.

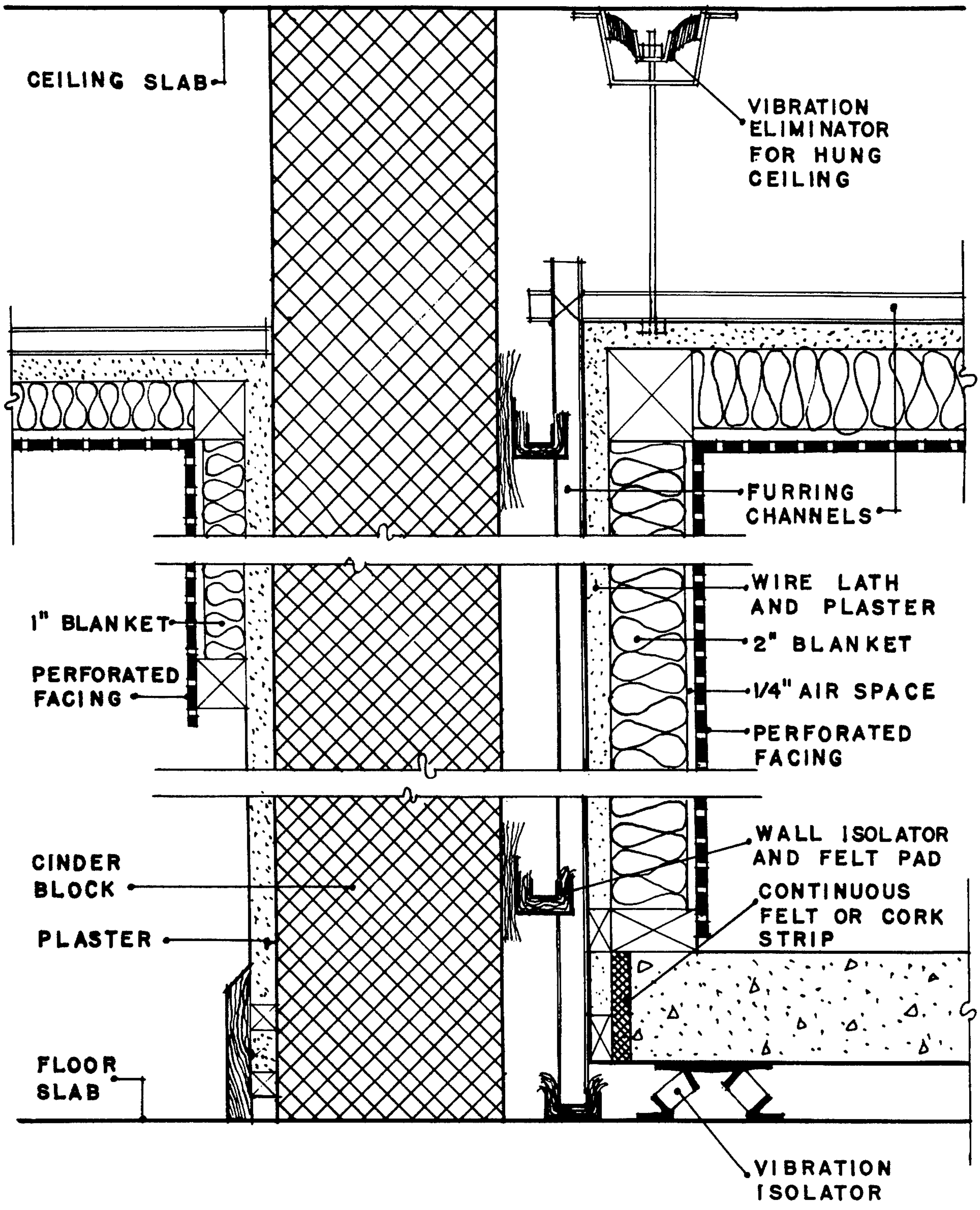
A detail of the wall, floor and ceiling constructions of the dead rooms, indicated as Rooms A and D of Fig. 13.4, is given in Fig. 13.5. The overall construction is that of a "floating" room within a room. In new construction, of course, the floor levels in these floated rooms could be adjusted to match the other floors in the building.

Several details in Fig. 13.5 are worthy of further discussion. The grounded copper shield screen surrounding the floating room is necessary to make the electronic sound measuring apparatus free from outside interference sources, such as fluorescent lights and electro-therapy equipment.

The elimination of structure-borne noise has been provided for in the resilient mounting of the reinforced concrete sub-base for the floating room. Either springs damped by mounting on felt or rubber-in-shear vibration isolators may be used. Their resonant frequency should be as low as possible, certainly not exceeding 5 cps.

Figure 13.5

Floating room which provides complete sound isolation and freedom from vibration.


CEILING SLAB
VIBRATION ELIMINATOR FOR HUNG CEILING
FURRING CHANNELS
WIRE LATH AND PLASTER
1" BLANKET
2" BLANKET
PERFORATED FACING
1/4" AIR SPACE
PERFORATED FACING
WALL ISOLATOR AND FELT PAD
CINDER BLOCK
CONTINUOUS FELT OR CORK STRIP
PLASTER
FLOOR SLAB
VIBRATION ISOLATOR

A subway grid or rigid perforated grill mounted on 4 in. T-sections is specified as the floor of the floating room. The grill is necessary to support the occupants while the open work is necessary acoustically to allow absorption of sound by the floor treatment.

Mineral wool covered with flame proof cloth of light weave is specified for the wall and ceiling covering.

Windows for observation purposes (not shown in detail) should be tightly mounted in rubber or neoprene gaskets while the doors should also be tight fitting. The doors should be covered with the copper shielding and provision should be made for continuous contact of this shielding with the shielding of the room.

All electrical passages into the room should be tightly sealed against sound leaks through the pipes. One inch of felt inside the cable box can be specified to prevent outside noise from being transmitted along the cable conduits.

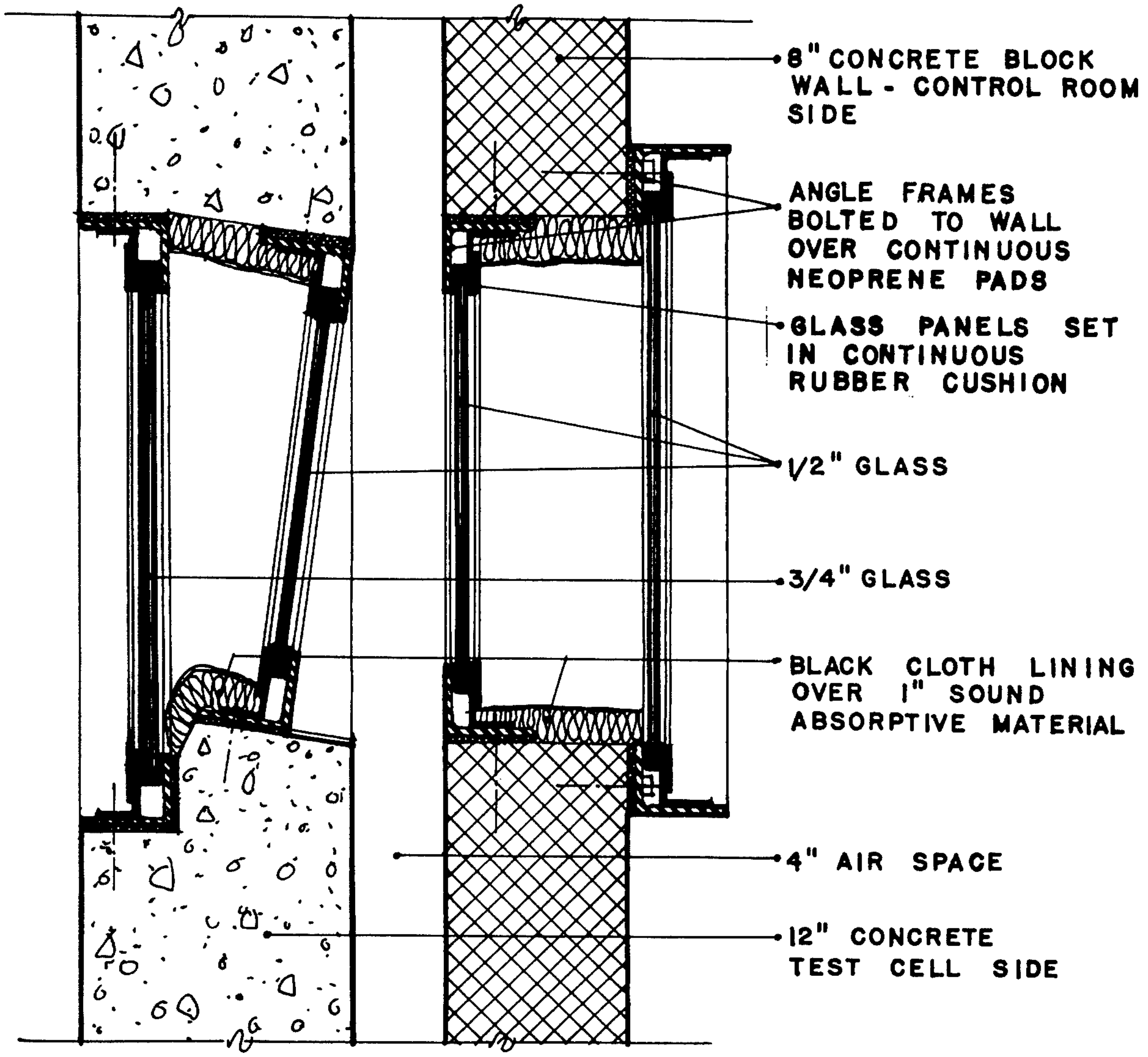
Figure 13.6 shows details of the Control Room walls. These walls are of lighter construction, but the comments made for the Anechoic Rooms are also applicable here.

13.7 Sealing of Windows and Doors

The optimum transmission loss in windows and doors can be obtained only with completely airtight sealing. Figures 13.7 and 13.8 show details of some seals which have been employed successfully for windows and doors respectively. It should be emphasized that proper thickness, number, and spacing of the glass panels in the observation windows should be determined to meet the specific transmission loss requirements of any particular problem (see reference (6), Chapter 11).

Figure 13.6

Details of control room wall



8" CONCRETE BLOCK WALL - CONTROL ROOM SIDE
ANGLE FRAMES BOLTED TO WALL OVER CONTINUOUS NEOPRENE PADS
GLASS PANELS SET IN CONTINUOUS RUBBER CUSHION
1/2" GLASS
3/4" GLASS
BLACK CLOTH LINING OVER 1" SOUND ABSORPTIVE MATERIAL
4" AIR SPACE
12" CONCRETE TEST CELL SIDE

Figure 13.7

Window seals, used to minimize the sound transmitted around the edges of a window.

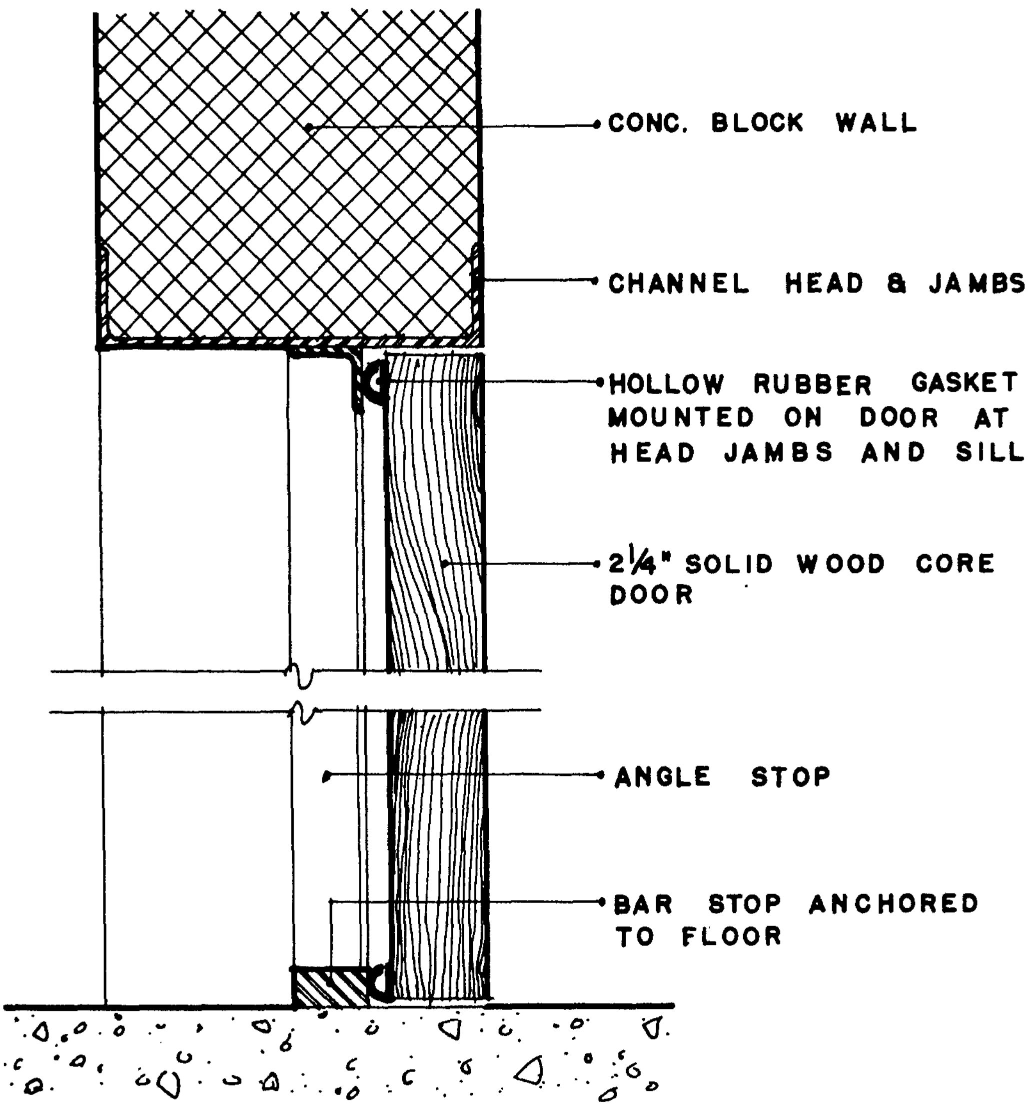

CONC. BLOCK WALL
CHANNEL HEAD & JAMBS
HOLLOW RUBBER GASKET MOUNTED ON DOOR AT HEAD JAMBS AND SILL
2¼" SOLID WOOD CORE DOOR
ANGLE STOP
BAR STOP ANCHORED TO FLOOR

Figure 13.8

Door seals used to minimize the sound transmitted through the cracks around a door.

References

(1) Beranek, L. L., "Developments in Studio Design", Proc. Inst. Radio Eng. 38 470-4 (1950)

CHAPTER 14

EVALUATION OF SOUND-CONTROL INSTALLATIONS

14.1 Transmission Loss and Noise Reduction

This chapter will discuss methods of evaluating sound-control systems intended for reduction of noise levels. A sound-control system, in this sense, is an acoustical attenuator which is employed to reduce the sound pressure level at a certain point of observation, when a given source is in operation at another point.

Two different kinds of evaluation of a sound-control system are usually of interest. The first of these, which is expressed in terms of a transmission loss, is a measure of the reduction in total transmitted power which is obtained by the use of a given acoustical structure (muffler, parallel-baffle system, etc.); the second kind which is expressed in terms of the noise reduction is a measure of the reduction in sound pressure level obtained at a designated location when sound-control treatment is installed.

These two evaluations are quite dissimilar. This can be made evident by considering a specific case; e.g. the problem of reducing sound pressure levels which exist at the windows of an office building when a jet-engine test is in progress in the neighborhood. Suppose that initially the tests are performed outdoors. To reduce the sound levels existing at the windows of the office building, changes are made as indicated in the listing below, where the decibel values represent the reductions of SPL obtained in the 1200-2400 cps frequency band.

(1) The distance between the source and the office building is doubled (6 db).

(2) The engine under test is enclosed in a building in such a way that practically all radiated sound must escape through a horizontally directed opening at the end of a duct; this opening is directed at an angle of 90° from the building. Therefore a reduction in SPL, due to directivity, is introduced (18 db).

(3) An isolation wall is erected between the opening and the building. The wall produces a reduction in level (as measured at street level) of 15 db.

(4) The duct is lined and parallel baffles are installed, with a resulting reduction in level of 40 db.

The total reduction in sound pressure level, for the particular location which is of interest, is the sum of the four values above, or 79 db. This is the noise reduction introduced by the sound control measures described. The first three corrective steps, however, involve no power reduction, but only a redirection of the radiation and a spreading of the acoustical energy over a greater area. Only the fourth step in the acoustical treatment acts to reduce the power radiated from the source.

It will be desirable in most cases to have the results of both evaluations. The noise reduction must be measured in order to ascertain whether the required reduction in SPL is obtained at those particular locations where the noise level is objectionable. The transmission loss, however, is required as a measure of the effectiveness of any acoustic absorbing element (muffler, lined duct, parallel baffles, etc.). Since a large part of the expense of noise-reduction measures is incurred in the construction of absorbing elements, it is desirable to evaluate these specifically, both as a check on the actual performance of the installed system, and as a guide for future engineering work. The transmission loss of an acoustical component is a unique measure of its behavior and one which, over a wide range of conditions, is independent of the system in which the component is used.

In the following sections, the two evaluation quantities are defined more rigorously, and some specific methods of measurement are considered.

14.2 Measurement of Transmission Loss

There are several possible alternatives when defining a measure of the power loss within an acoustical component. The transmission loss, as defined below, has been selected as being most suited to the situations which arise in this handbook.

The transmission loss, in decibels, of an acoustical component, (TL) is equal to 10 $\log_{10}(P_2/P_1)$, where P_1 is the power delivered to the structure from the source and P_2 is the power which emerges from the structure in such a way as to contribute to the externally observed sound pressure.

The significance of the definition, as it applies to a situation which is commonly encountered, may be explained with reference to Fig. 14.1. Here the source, which produces power P_o in a certain frequency band, is located in an enclosure from which sound can escape only through a duct having an effective transmission loss TL. Let A_d be the area of the duct opening in sq ft. Let A_r be the number of units of absorption provided by the walls of the enclosure including the duct opening.

In the simplest case, the walls of the enclosure except for the duct opening have negligible absorption. Then all power produced by the source must enter the duct so that P_1 is equal to P_o, and the following simple relations holds:

$$PWL_o - PWL_2 = TL \qquad (14.1)$$

Here PWL_o is the power level of the source and PWL_2 is the power level of the sound which escapes as found by external measurements. Both values must relate to the same frequency band.

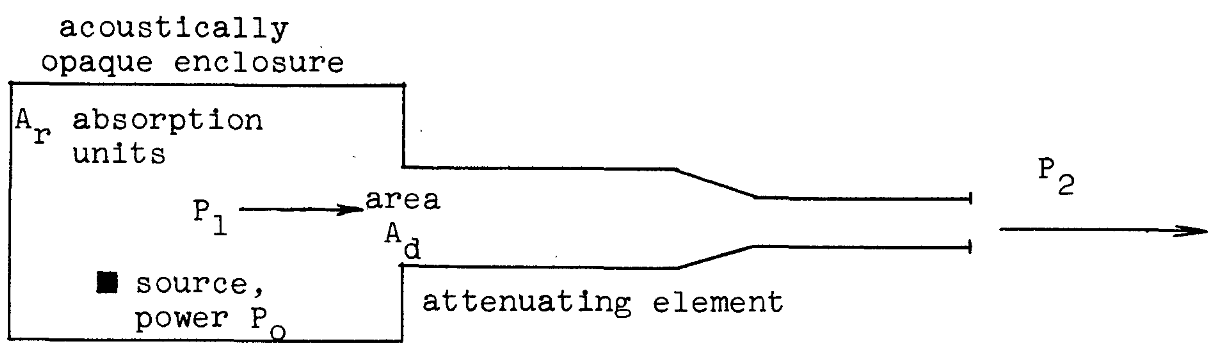


Figure 14.1

Noise source located in an enclosure which is coupled to a duct.

When the room walls have appreciable absorption, a rigorous description of the situation is very difficult, but a crude correction for power lost to the room walls may be made by the use of the simple diffuse sound field assumptions of architectual acoustics. According to these assumptions, the power incident upon the duct entrance is given approximately by $P_1 = P_o (A_d / A_r)$, and the relation expressed by Eq. (14.2) applies

$$PWL_2 = PWL_o - TL - 10 \log \left(\frac{A_r}{A_d}\right) \qquad (14.2)$$

In all cases the power delivered to the duct (P_1) is taken to be the duct area times the value of the isotropic sound intensity in the room. Therefore P_1 is the incident power. Since some sound energy may be reflected back into the enclosure, P_1 is not necessarily equal to the net power delivered to the duct. This distinction is analogous to one which is observed in discussion of the transmission loss of panels. Thus the transmission loss as defined above is a direct generalization of the quantity conventionally used to describe structural transmission of airborne sound.

These relations serve both to define the transmission loss and to show how known values of TL can be used to relate true source power level to the apparent power level as found by external measurement when the attenuator is in place.

Since the TL is found by measurement of power levels, any applicable method described in Chapter 6 for the measurement of power level may be used as a basis for a transmission loss evaluation. Usually the method depends upon the summing up of power values for a number of portions of an area through which all the sound must pass. In any method of evaluating the TL, it is essential that the external sound measured actually comes from the attenuating structure which is being evaluated. The sound energy at each measurement station, coming from the desired point of radiation and through the structure under investigation must be greater by at least 10 db than that arriving by all other transmission paths, either from the desired source or from interfering sources. In particular, this restriction holds for those measurement stations which account for 90 percent or more of the radiated power. A few specific methods for TL measurement will be considered in the following sections.

1. The method of <u>distance measurements</u>. "Distant" refers to points separated from the radiating surface area by at least several times its largest dimension. With the acoustic attenuator in place, a spectrum of SPL is measured at a group of "distant" microphone positions which are selected to allow computation of the power level spectrum (PWL_2) of the radiated sound (see Chapter 3). The power level spectrum of the source alone (PWL_o) is also evaluated by any of the methods of Chapter 6. The evaluation of PWL_o is best achieved by operating the source outdoors or in an enclosure which allows the sound energy to escape through untreated openings while an external sound survey is made. The TL of the attenuator is then found using Eq. (14.1) or Eq. (14.2).

1a. <u>Special Case</u>: <u>The method of distant measurements for the determination of the insertion difference of an attenuator</u>. A special case of Method 1 may be termed a "before and after" survey. First, a distant point survey is made with a given source operating in a partial enclosure but with no acoustic treatment present at the opening or within the enclosure. From this survey the effective source power level (PWL_o) is found. After the acoustic attenuator has been installed in the opening of the enclosure, the same survey is repeated, with the original sound source in operation. From the second survey the power level of the radiated sound is found (PWL_2). The TL is found from Eq. (14.1).

Method 1 (and particularly the modification 1a) has the virtue of directness and is accurate in principle. It is desirable, in Method 1a, to take auxiliary readings if possible with a microphone placed in a standard position a few inches from the source, in order to establish that the same source output is maintained during both the initial and final sound surveys. A correction may be added to the results if necessary to compensate for any change in the source characteristics. The practical difficulties associated with Method (1) may be summarized as follows: (a) often it is not feasible to place a survey microphone in all positions required for distant-measurement evaluation of power level (in particular, impractical elevated positions may be required); (b) in some cases sound is radiated at comparable power levels from two attenuator openings simultaneously (for example, from both intake and exhaust ducts); (c) the reduced sound radiation with the attenuator in place may be too small to be measured at any appreciable distance in the presence of neighboring noise sources.

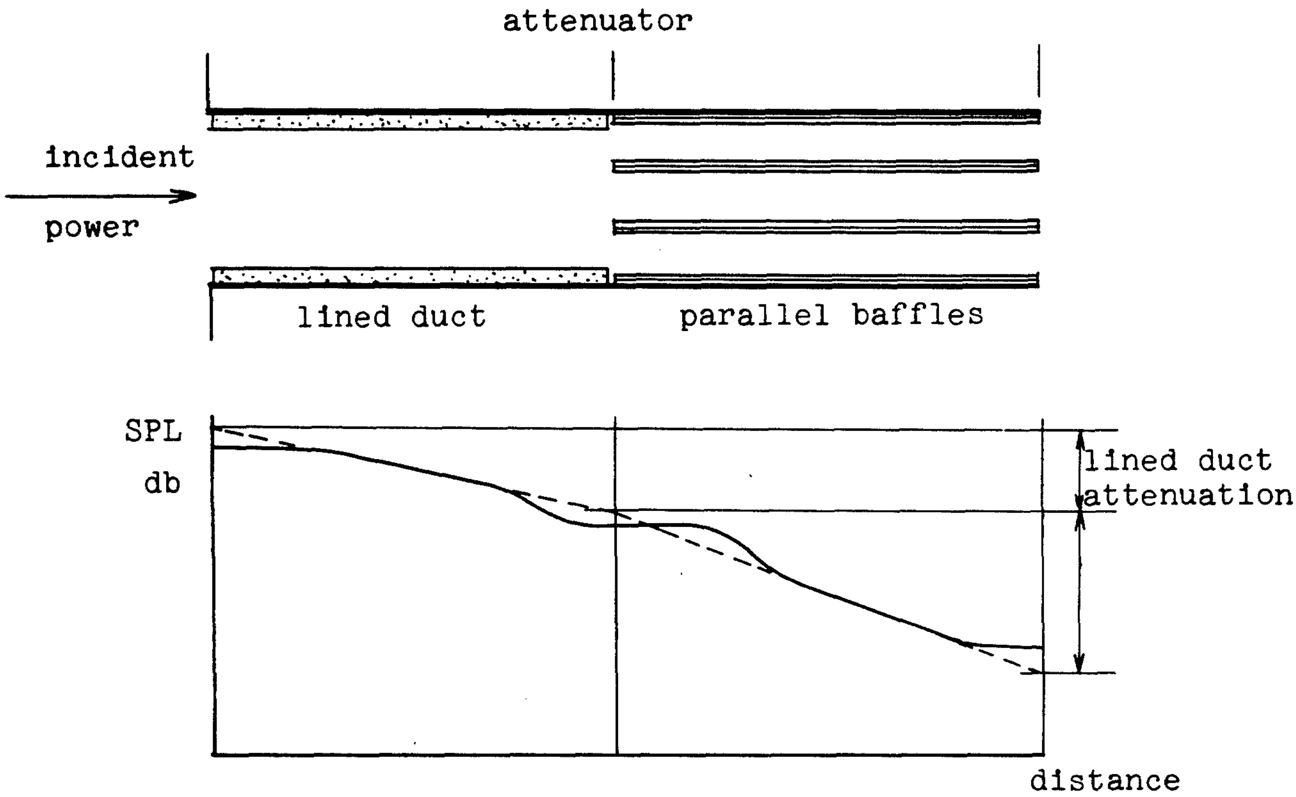
2. The method of adjacent measurements. The method of adjacent measurements differs from the distant measurement method only in that the plane of the radiating opening, rather than a distant area, is selected as the survey surface for measurement of the radiated power level (PWL_2). The source power level may be found by any convenient method, as before. If the source power level is evaluated by an adjacent-measurement survey in the plane of the enclosure opening before installation of the acoustic attenuator, Method 2 becomes a "before and after" method. The TL can then be calculated from Eq. (14.1) or Eq. (14.2).

The method of adjacent measurements would be rigorously accurate in principle if the data could be obtained using a true sound intensity meter. Since such intensity meters are not available in a practical form it is necessary to compute the power from sound pressure readings as indicated in Chapter 3. This computation in which it is assumed that the sound intensity is equal to $p^2/\rho c$, is valid only when applied to plane-waves or spherical waves at large distances from the source. When these conditions are not realized, the power actually associated with a given set of pressure readings is less than the calculation indicates. In spite of this difficulty, the adjacent method is useful in many cases. Results obtained this way, however, should be designated as tentative or approximate.

3. The method of end differences. Two multiple-point surveys of SPL are carried out with the attenuator in place, and with a given source in operation, one survey covering the source or input plane of the attenuating structure and the other survey covering the output plane. If it can be assumed that there is no significant reflection at the entrance of the attenuator, the power (P_1) entering the attenuator may be computed from the relation:

$$P_1 = S(p^2)_{av}/\rho c^2 \qquad (14.3)$$

where S is the area of the input plane opening, $(p^2)_{av}$ is the pressure squared averaged over the opening, and ρc is the specific impedance of air under the conditions prevailing at the entrance. The same relation is used to compute (P_2) the power leaving the attenuator at the output end. The TL is given by Eq. (14.4) as:


attenuator
incident
power
lined duct
parallel baffles
SPL
db
lined duct
attenuation
distance

$$TL = 10 \log (P_1/P_2) \qquad (14.4)$$

The end-difference method could be made highly accurate if it were possible to make direct intensity measurements. Since the method is based instead on making pressure measurements and reducing the data with Eq. (14.3), the error described in connection with Method 2 is again involved. In addition, the assumption that the attenuator is non-reflective is often unjustified; for example, a significant reflection is usually introduced by a bend. The end-difference method affords a convenient means for investigating a treatment already in position. If the attenuator is actually non-reflective, the end-difference method gives results of about the same accuracy as the adjacent-measurement method. The results obtained with either method should be regarded as only approximate.

4. The traversing method. In this method the sound pressure level is measured at a number of positions along the attenuator, by means of a microphone which is placed at various points within the attenuator as required. The data for each frequency (or frequency band) are presented as a plot of SPL as a function of distance from the input plane. Fig. 14.2 shows the appearance of the resulting plot for a case where the attenuator contains two different treatments. A straight line of "best-fit" is drawn on the SPL vs distance graph for each section of the treatment as shown in Fig. 14.2. It is sometimes necessary to disregard the behavior of the plot near the ends of the treatment in drawing this straight line. The total attenuation for each section is obtained by extrapolating the straight line to the full length of the section, as indicated in the figure. The TL of the entire structure is the sum of the total attenuations of the sections, plus 10 log (input area/output area). The attenuation constant for each section (in db/unit length) is given by the slope of the straight line for that section.

The slope of the SPL as determined in the traversing method is, in effect, the attenuation constant for an infinitely long treatment. This attenuation constant is then used to find the total loss in the existing, finite length of treatment. The

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Figure 14.2

Investigation of attenuator performance by means of the traversing method.

loss found in this way is nearly equal to the TL if the following two conditions are realized: (a) The energy entering the attenuator is not reflected appreciably (b) The energy reaching the far end of the attenuator structure is radiated without appreciable reflection. It may be assumed that these conditions are realized at frequencies such that the perimeter of the duct is greater than the wave length of sound. The method is consequently nearly correct at all audio frequencies when the attenuator is as large as a 10 ft x 10 ft square duct.

The computation of intensity from sound pressure measurements which was mentioned as a source of error in Methods 2 and 3, is substantially eliminated in the traversing method by the method of treating the data to obtain an effective rate of attenuation by relative rather than absolute measurements.

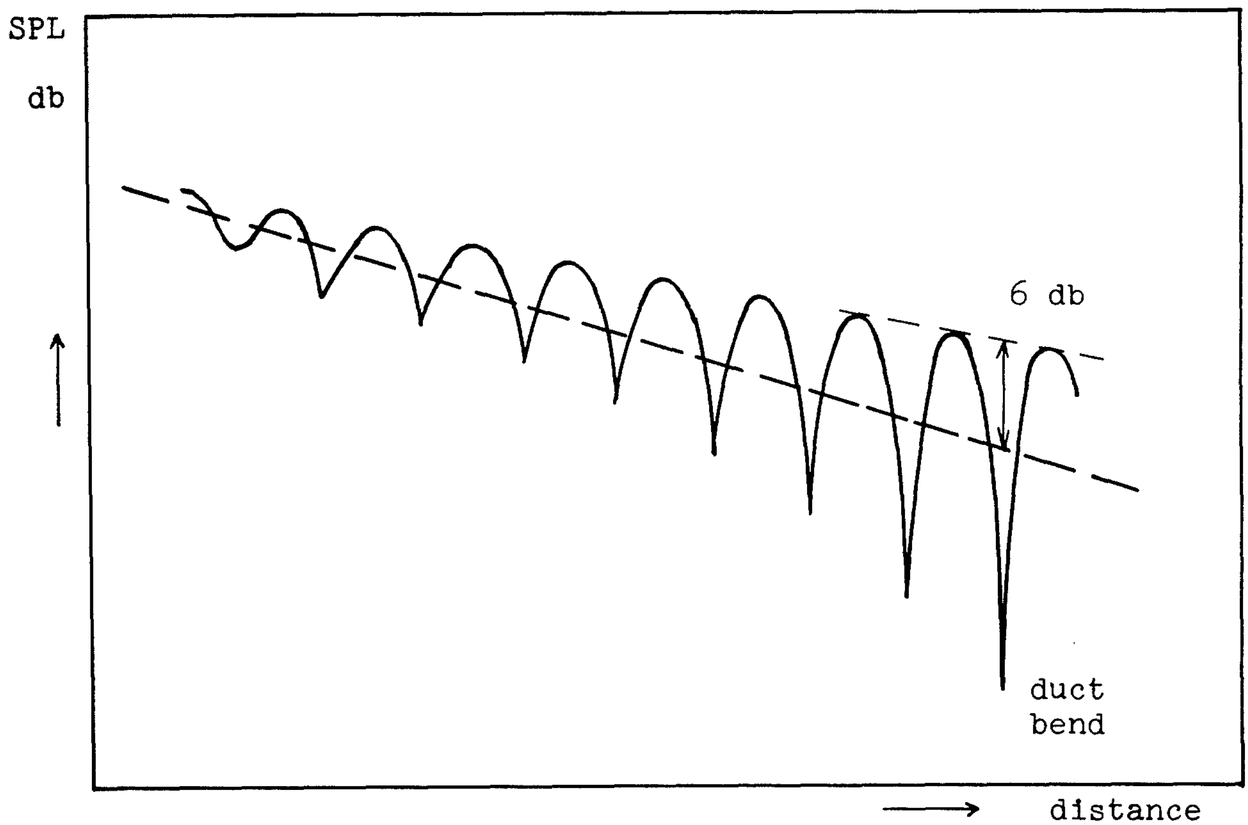


Figure 14.3

Result of applying the traversing method to a duct in which reflection is taking place. Such a result will be obtained only when the measurement is made with a pure tone or a narrow band of noise

When significant reflections take place within the attenuating structure and when the traversing measurement is done with a single pure tone or with a very narrow band of noise, the plot of SPL vs distance may have the appearance shown in Fig. 14.3. The fluctuations are due to standing-waves, produced in this case by reflections from a bend. The presence of these standing-waves does not invalidate the measurement technique if the standing-wave variations in SPL at the source end of the structure, and in the final section which opens to the air, are small, less than 5 db peak to peak. The correct method for drawing an average straight line when standing waves are present is indicated in Fig. 14.3. Where the pressure variations are less than 5 db, the line passes midway between the maxima and the minima (lefthand portion of graph). As the amplitude of the variations increases, the line is kept nearer to the locus of the maxima. When the variations exceed 25 db total, the average line lies 6 db below the maxima. In practice the severe standing-wave effect shown in Fig. 14.3 will seldom be found, and it is usually satisfactory to draw a line midway between the maxima and minima.

The traversing method has the particular advantage of being adapted to measurements in the presence of high external noise levels. Since the microphone is placed within the attenuator, the attenuator itself reduces the effect of external noise sources upon the readings, except when the microphone is near the exit. The effect of external noise on the readings taken near the exit is to cause these readings to depart from the straight line drawn through the intermediate points; this effect is ignored in the normal method of interpreting the data.

The traversing method is also suitable for the evaluation of attenuating elements which are localized in the structure, such as single resonators and duct bends. For any frequency at which the localized attenuator is effective, the plot of SPL versus distance assumes nearly constant values over a range of distances on either side of the unit, but has a rapid change or a discontinuity in the region of the attenuating element. The difference between the averaged values of the SPL on the two sides of the attenuating element is the TL due to the element. In certain ducts it may be impossible to use the traversing method because of baffles, valves, flow regulating devices, or other mechanical equipment which may be present.

Recommendations for choice of method. The method chosen for a given application may be dictated by limitations on available time or personnel; in this case it may be necessary to use Method 2 or Method 3. When a compromise does not have to be made on the basis of practicality, the methods may be ranked as shown below, where the preferred method is listed first.

I. The method of distant measurements (Method 1) (particularly the "before and after", or insertion difference modification).

II. The traversing method (Method 4).

III. The method of adjacent measurements (Method 2).

IV. The method of end-differences. (Method 3).

If measurements must be made on an attenuator which is already in place, and the power level spectrum of the source is not known, only the traversing method and the method of end-differences are applicable.

The traversing method has, in addition to its application in finding the transmission loss, value as a diagnostic tool. As a first example, the existence of flanking transmission is indicated by a leveling off of the plot of SPL versus distance in the regions of the treatment farthest from the source. The statement has been made in commercial practice that no useful attenuation is contributed by extending an attenuating treatment to a length greater than 12 ft. Investigation, by the traversing method, of the structures which gave rise to this statement showed the existence of flanking transmission. Installation of vibration breaks and of adequate duct walls eliminated flanking transmission and allowed the full, calculated attenuation to be realized. Total attenuations as large as 80 db have been observed under properly controlled conditions.

As a second example of the diagnostic value of the traversing method, a plot of SPL versus distance which rises toward the output end of the attenuator indicates the presence of external noise levels greater than those which would be produced by sound transmitted through the attenuator. This additional noise component may be the result of another source, or the result of bypass transmission, through an external path, from the original source which is to be isolated. When this

condition exists, transmission loss values obtained by the end-difference method will be badly in error.

This list of methods of measurements is not exhaustive. Many modifications, or combinations of the methods listed, may be worked out to meet particular requirements. In particular, the performance of an attenuation duct connecting two "live rooms" (for example, an air-conditioning duct) may be evaluated by applying the principles developed in Chapter 11 in connection with the transmission loss of panels. The definition which is given in the present chapter for transmission loss agrees with the definition previously stated for the transmission loss for a panel. If the attenuation of an air-conditioning duct is large, it is usually necessary to apply the traversing method.

14.3 Measurement of the Noise Reduction

The distinction between transmission loss and noise reduction was discussed in Sec. 14.1. On the basis of that discussion, the definition which follows gives the specific meaning of noise reduction in the present chapter.

The noise reduction (NR) is the decrease, attributable to a designated set of noise-control operations, of the sound pressure level at a specified observation point.

Evidently the noise reduction can be defined only in connection with a specific combination of noise-control measures and at a definite observation point. In general the effective noise reduction is the sum of the TL due to any attenuating structures present plus the additional reductions in sound level which are due to such causes as directivity, the spreading of sound, or outdoor atmospheric absorption.

Measurement of noise reduction as defined above depends inherently on a "before and after" technique. The difference in the SPL at a designated observation point, before and after the installation of specific sound-control measures, is directly the NR. If the effective power level of the sound source is altered in the interval between the two measurements, and if this alteration is not attributable to the sound-control procedures, an appropriate correction should be made. Changes in the effective power level of the source can be determined from the indication of a pressure microphone placed in a standard position near the source.

Since NR in some cases includes the attenuating effects of outdoor sound propagation, results at distances greater than a few hundred feet from the source will depend upon atmospheric conditions, and upon the surface condition of the ground. Therefore any available pertinent information concerning these outdoor conditions should be included as a part of the information reported on noise reduction at large distances. A noise-control problem of large scope will warrant continued observations to include all representative meteorological conditions. If the sound-control measures do not change the directivity of the radiation from the source, it may be assumed that the variation of sound level at the observation point, due to changing outdoor conditions, is the same before and after the sound-control installation. If the source directivity is changed, the relative influence of atmospheric conditions may shift to a considerable degree (see Secs. 12.8 and 12.10). In either case a reliable value for the NR will be obtained only if similar outdoor conditions exist for the initial and final sound level measurements.

When the acoustical designer has adequate advance knowledge of the expected power level spectrum for a noise source, it is possible to design the noise-control system at an early stage in the development of plans. The noise-control system may then be installed at the same time as the equipment which makes up the noise source with the result that unacceptably high noise levels are never experienced by surrounding areas. When this is true, the total noise reduction as discussed is not measurable, since there is no "before and after" comparison. It is often preferable in this case to use an easily calculated reference state in finding the total noise reduction. For this purpose the concept of noise reduction relative to an isotropic source is a useful one.

The noise reduction relative to an isotropic source is the difference between the SPL which would exist at a designated point of the sound energy diverged in a uniform spherical fashion from the source, and the actual SPL at that point. Thus, if PWL_o is the power level of the source in a given frequency band, and if SPL is the sound pressure level actually observed at the specified location in this frequency band,

$$\left[\begin{array}{l}\text{NR (relative to} \\ \text{isotropic} \\ \text{source)}\end{array}\right] = PWL_o - 10 \log (4\pi r^2) - SPL \qquad (14.5)$$

where r is the distance in feet from the source to the point of observation. Note that noise reduction defined in this manner can assume negative values when the TL is small, because the effects of directivity, atmospheric refraction, and ground reflection can increase the observed levels in specific directions over the isotropic distribution value.

As explained under the definition of noise reduction in Chapter 2, this term is used not only to describe the difference in SPL at a single point, before and after the installation of noise-control structures, but also to describe the difference in noise levels between two locations, at a single time. For example, "noise reduction" may be used to describe the difference between the SPL observed at the same time in an adjacent room, the readings in each case being attributable to the same sources. In general no special techniques are required to evaluate the noise reduction in this sense. Therefore noise reduction as a difference in levels at different locations is not considered in this chapter.

14.4 Sound Control Evaluation as a Function of Frequency

The effect of frequency must be considered in all evaluation measurements. This applies to both transmission loss and noise reduction determinations.

Noise Reduction. Since sound-control frequency requirements are ordinarily expressed in terms of sound level within octave bands (Chapter 10), it is natural to express NR as a function of frequency in the same way. Therefore, measurements used for evaluating noise reduction are normally made with an octave-band sound analyzer. More detailed information can be obtained by use of an analyzer with 1/3-octave bands, or a narrow-band analyzer. The extra effort required to obtain more detailed information is usually justified only in special cases. As an example, use of a narrow-band analyzer may be justified in evaluating the reduction of propeller noise. Since propeller noise consists largely of discrete frequency components, it is possible to adjust a narrow-band analyzer to the frequencies of these components and thus to reject interfering signals from other sources. Even when measurement using a narrow frequency band is desirable, the data should be converted to octave-band values for comparison with criteria for acceptable noise levels, (see Volume II).

Transmission Loss. Since TL is a measure of the performance of a carefully designed structure, it is desirable to know the variation of this quantity with frequency in as much detail as possible. Thus it is desirable to measure transmission loss with pure tones.

In practice, measurements with pure tone sources are not usually accomplished, because (a) pure tone sources of sufficient power are not readily available; (b) the interpretation of the data for engineering purposes is more difficult when pure tones are used than when bands of noise are used, because standing-wave effects are more pronounced with pure tones.

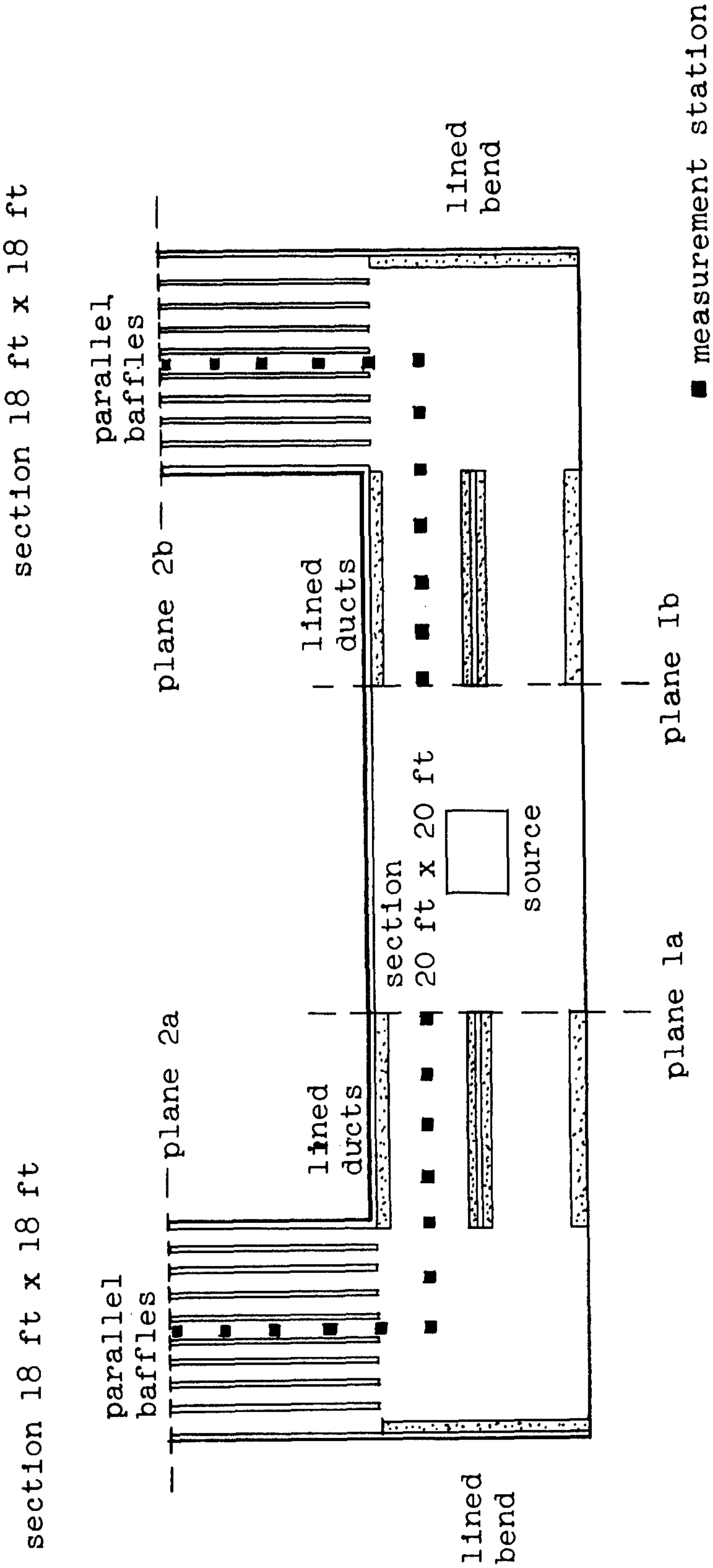
When pure tone measurements are not practicable, attenuation measurements, or other measurements leading to TL, should be performed with the narrowest possible frequency bands, using a noise source. Ordinarily, the simplest way is to use the actual noise source (aircraft engine, etc.) as the test signal generator, and to use an electrical sound analyzer to examine the received signal. A narrow-band analyzer is preferred; next in preference is the 1/3-octave band analyzer; and finally, the octave-band analyzer. The octave-band analyzer is used for most field measurements, because it requires the minimum operating time. Use of analyzers having narrower frequency bands, and therefore requiring more time for adjustment, is often ruled out because of limitations on the permissible operating time of the sound source.

Figs. 12.4 and 12.5 show the same data for TL as a function of frequency, but expressed first on an octave-band plot (Fig. 12.4), and then on a pure-tone frequency scale (Fig. 12.5). The greater detail obtained in the second presentation is apparent. These results are reduced to values of the attenuation constant in db/ft.

14.5 Numerical Example of Evaluation

The hypothetical structure shown in Fig. 14.4 will be taken as a basis for a numerical example of an evaluation survey. This structure represents a possible arrangement of an acoustically treated test cell for operation of reciprocating engines and propellers. For generality, it is assumed that the system is not symmetrical and that the two ends must be investigated separately. The values used in the example are chosen for illustrative purposes only and are not necessarily representative of actual practice. The detailed discussion will deal only with values for the 600-1200 cps frequency band, in order to shorten the presentation.

The possibility of a survey by the distant-measurement method is first investigated. This method is ruled out by the consideration that the axis of symmetry of each radiating


section 18 ft x 18 ft
parallel baffles
plane 2b
lined ducts
lined bend
measurement station
plane 1b
section 20 ft x 20 ft
source
plane 1a
plane 2a
lined ducts
section 18 ft x 18 ft
parallel baffles
lined bend

opening is vertical, so that it would be necessary in a distant survey to include microphone positions elevated by as much as 100 ft from the ground.

It is decided that the evaluation will be carried out by a combination of the end-difference method and the traversing method. The resulting information will give both TL and NR relative to an isotropic source.

The first series of measurements consists of SPL surveys of the planes marked 1a, 1b, 2a, and 2b. A few representative locations in each plane (corner areas and central areas) are investigated. Variations of SPL in each plane are found not to exceed ± 2 db. The average values accordingly are as follows:

Plane 1a	120 db
Plane 2a	82 db
Plane 1b	118 db
Plane 2b	83 db

An approximate indication of the TL values may now be obtained by applying area corrections to the above data. Thus, for the left-hand attenuator, the transmission loss is

$$TL_a = 120 - 82 + 10 \log \left(\frac{20^2}{18^2}\right) = 39 \text{ db}$$

Similarly,

$$TL_b = 36 \text{ db}$$

To secure more accurate results, a series of measurements is now made by traversing each attenuator. The traversing paths are laid out so as to join the end planes at points where the SPL has approximately the average value in each plane. The measurement paths are indicated in Fig. 14.4. Data obtained in the traversing measurements are treated in the manner described in Sec. 14.2. The total attenuations are found to be 34 db for the left-hand section and 32 db for the right-hand section. The TL is obtained by adding, to each of these values, $10 \log [20/18]^2$,

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Figure 14.4

Hypothetical structure consisting of a test cell with an intake and exhaust muffler, used as the basis for an example of an evaluation survey.

which is 1 db. Thus, by the more accurate method,

$$TL_a = 35 \text{ db}$$

$$TL_b = 33 \text{ db}$$

These results are somewhat lower than those obtained by the end-difference method.

The sound level at a certain location 500 ft distant from the source is of particular interest. A measurement at this location shows that the SPL is 36 db. It should be noted here that level as low as 36 db in the 600-1200 cps band can be measured reliably only when there is no nearby heavy traffic or other interfering source.

It is now desired to find the power level of the source (PWL_o) and to compute the NR relative to an isotropic source. The SPL values obtained at the external openings are assumed to give a more reliable basis for finding the power level of the source than the values obtained within the enclosure, because of absence of reflections at the external openings. For the left-hand side, the average SPL at the external opening is 82 db. If there were no power loss in the structure the level would be (82 + 35) db, or 117 db. The sound pressure corresponding to a SPL of 117 db is

$$p = 0.0002 \frac{\text{dynes}}{\text{cm}^2} \times 10^{117/20} = 141 \text{ dynes/cm}^2$$

The total power corresponding to this pressure, for sound transmission through an 18 ft x 18 ft area, (3.01×10^5 cm^2) is

$$\text{Power} = \text{Area} \times p^2 \times 10^{-7} \quad /\rho c \quad (\text{watts})$$

$$= 14.6 \text{ watts}$$

Similarly, the power entering section b, as computed by correcting the externally observed SPL, is 11.5 watts.

The total effective acoustic power of the source is, therefore, 26.1 watts which gives a power level in decibels of

$$PWL_o = 10 \log [26.1/(0.9 \times 10^{-13})] = 145 \text{ db}$$

Eq. (14.5) may now be used to find the noise reduction relative to an isotropic source. The result is

$$NR = 145 - 10 \log (4\pi \times 500^2) - 36$$

$$= 44 \text{ db}$$

The NR, relative to an isotropic source, exceeds the average TL by (44 - 34) or 10 db. This additional reduction of 10 db represents all effects due to preferential distribution of the sound radiated from the openings. The specified point of observation, at ground level, is at approximately 90° from the axis of each radiating area. The directivity index for a very large stack opening at 90° is about 20 db, indicating that the effective noise reduction secured is 10 db less than would be predicted from the source directivity. The probable explanations of this difference are:

(1) Reflection from the ground increases the observed level. Ordinarily this effect does not exceed 3 db.

(2) Refraction of sound is occuring as a result of gradients of velocity and temperature in the air. If one of the openings of the enclosure is discharging warm exhaust gas, this explanation is especially plausible. The upward velocity of the air above the opening and the increased temperature, both existing in a restricted region, would have the effect of refracting vertically radiated sound.

The example discussed above is a case in which a reasonably accurate measurement of TL can be made. It should be pointed out that in some instances this measurement cannot be performed accurately with ordinary facilities. For example, suppose that the vertically directed openings of Fig. 14.4 were modified by installing above each a flat roof section, lined on the lower side with absorbing material. The resulting arrangement is shown in Fig. 14.5.

It is very difficult to determine the total power radiated by this configuration. Adjacent readings of SPL, taken with the microphone in position A, will give too large a value for the radiated power. The reason for this is that a portion of the pressure variation at the microphone is due to sound which is not radiated, but which is absorbed on the lower side of the roof. An adequate distant-point survey would require a number of elevated microphone positions which cannot be reached with microphone supports ordinarily available. Traversing the attenuator

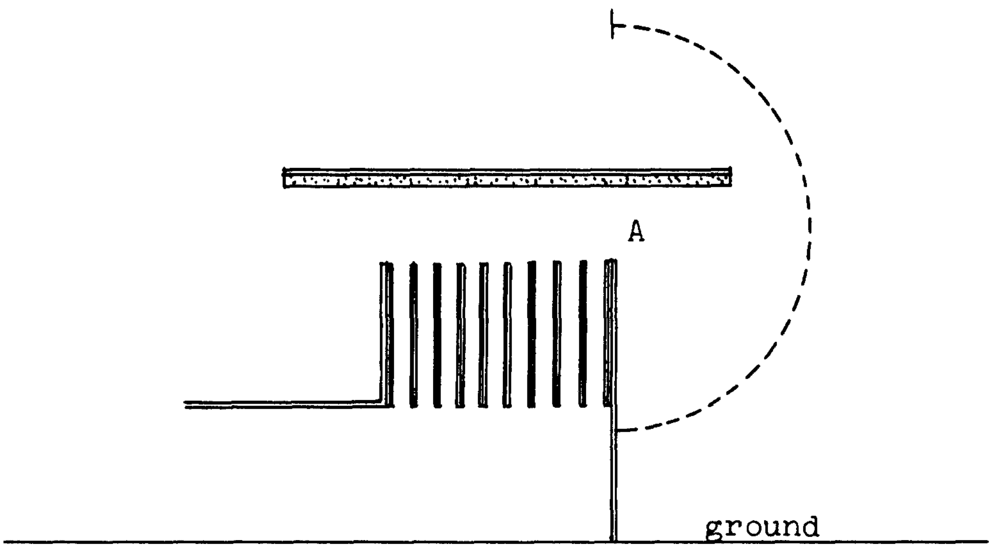

Figure 14.5

A more complicated muffler opening, for which the radiated power is difficult to measure.

will not give accurate results, because reflections occur at the roof (at least for low frequencies), and because the effective radiating area is not known.

In practice the engineer would have to be content with some approximate measurement of the radiated power. One approximate approach to the problem would be to make a survey along the arc shown by the broken line in Fig. 14.5, and to calculate the total power upon the assumption that the radiation is symmetrical about a vertical axis. The results would be inaccurate because of the effects of the several reflecting surfaces. Another approximate approach could be made by making an estimate of the directivity (or by measuring the directivity with a model) and then applying directivity corrections to SPL values measured near ground level at a large distance from the opening.

Fortunately a knowledge of the transmission loss is not essential to a functional evaluation of the complete noise-control system, although the performance of a specific muffler system can be described only by this quantity. The functional evaluation of the complete system is always indicated by the noise reduction at specified measurement points. The noise reduction can be directly measured in all cases which allow "before and after" surveys.